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POWER EFFICIENT HYDRAULIC SYSTEMS

Volume II

HARDWARE DEMONSTRATION PHASE

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Energy saving concepts for aircraft hydraulic systems were studied in a two-phase program. Task I was an investigation of methods and techniques to reduce overall hydraulic system power requirements by lowering system demands and increasing component efficiencies. Task II involved hardware demonstration tests on selected concepts.

Task I: Study Phase. A baseline hydraulic system for an advanced aircraft design was established. Twenty energy saving techniques were studied as candidates for application to the baseline vehicle. A global systems analysis approach was employed. The candidates were compared on the basis of total fuel consumption and six qualitative factors. Nine of the most promising techniques were applied to a "Target System". The target system had a 28% reduction in energy consumption and an 868 lb weight reduction over the baseline aircraft. The study made one conclusion clear: Don't add weight to save energy.

Task II: Hardware Demonstration Phase. Two techniques demonstrated for energy savings

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were control valves with overlap and dual pressure level systems. Tests were conducted on control valves, a servo actuator, dual pressure pumps, and a lightweight hydraulic system simulator. Valves with 0.002 in. overlap reduced system energy consumption 18% compared to using valves with zero lap. Operation at 4000 psi reduced system energy consumption 53% compared to operation at 8000 psi. Pressure level switching was accomplished with excellent results. Keywords →

FOREWORD

This report presents the results of the second phase of a two phase program to study and demonstrate methods and techniques to improve the operating efficiency of hydraulic systems in advanced Navy aircraft. The results of Task I (study phase) are presented in Volume I of this report. The results of Task II (demonstration phase) are presented herein.

The study phase consisted of the following:

- Determination of study methodology
- Definition of baseline vehicle
- Establishment of baseline hydraulic system
- Evaluation of candidate energy saving techniques
- Application of the most promising techniques to a target system
- Determination of weight and energy savings of the target system over the baseline.

The hardware demonstration phase consisted of the following:

- Design test parts
  - actuator modification
  - test fixture modification
- Procure demonstration hardware
  - Direct drive control valves and electronics
  - Dual pressure pumps
- Analyze test results
- Summarize results

TABLE OF CONTENTS

Section	Title	Page
	- - - VOLUME II - - HARDWARE DEMONSTRATION PHASE - - -	
	FOREWORD	i
	TABLE OF CONTENTS	ii
	LIST OF FIGURES	iv
	LIST OF TABLES	vi
1.0	HARDWARE DEMONSTRATION PHASE (TASK II)	
	1.1 APPROACH	
	1.1.1 Introduction	1
	1.1.2 Demonstration Hardware	
	1.1.2.1 Control Valves	1
	1.1.2.2 Servo Actuator	4
	1.1.2.3 Dual Pressure Pump	6
	1.1.2.4 LHS Simulator	7
	1.1.3 Tests Conducted	
	1.1.3.1 Control Valves	10
	1.1.3.2 Servo Actuator	11
	1.1.3.3 Dual Pressure Pump	12
	1.1.3.4 LHS Simulator	13
	1.1.4 Instrumentation	
	1.1.4.1 Hydraulic Test Bench	14
	1.1.4.2 Electronic Data Analysis System	14
	1.1.4.3 Transducers	17
	1.2 HARDWARE DEMONSTRATIONS	
	1.2.1 Control Valves	
	1.2.1.1 Procedure	17
	1.2.1.2 Results	22
	1.2.2 Servo Actuator	
	1.2.2.1 Procedure	28
	1.2.2.2 Results	39
	1.2.3 Dual Pressure Pump	
	1.2.3.1 Procedure	49
	1.2.3.2 Results	52
	1.2.4 LHS Simulator	
	1.2.4.1 Procedure	58
	1.2.4.2 Results	65
	1.3 DEMONSTRATION PHASE SUMMARY	
	1.3.1 Control Valves	80
	1.3.2 Servo Actuator	81
	1.3.3 Dual Pressure Pump	83
	1.3.4 LHS Simulator	84

TABLE OF CONTENTS (Continued)

Section	Title	Page
2.0	DEMONSTRATION PHASE CONCLUSIONS	85
3.0	RECOMMENDATIONS	86
	REFERENCES	88
	LIST OF ABBREVIATIONS AND SYMBOLS	89

APPENDICES

A	HARDWARE PROCUREMENT DOCUMENTS	91
B	CONTROL VALVE TEST DATA	111
C	SERVO ACTUATOR TEST DATA	127
D	DUAL PRESSURE PUMP TEST DATA	153
E	LHS SIMULATOR TEST DATA	159

LIST OF FIGURES

Figure No.	Title	Page
1	Bendix Direct Drive Servo Valve	3
2	E-Systems Direct Drive Servo Valve	3
3	Original Unit Horizontal Tail Actuator	5
4	Modified UHT Actuator with Bendix Valve	5
5	Modified UHT Actuator with E-Systems Valve	5
6	Vickers Dual Pressure Pump with Switching Valve	8
7	Lightweight Hydraulic System Simulator	9
8	8000 psi Hydraulic Test Bench	15
9	Electronic Data Analysis System	15
10	Block Diagram of Data Analysis System	16
11	Schematic Diagrams of Valve Test Setups	20
12	View of Flow Gain Test Setup	21
13	Flow Gain Data, Bendix Zero Lap Valve	23
14	Pressure Gain Data, E-Systems Overlapped Valve	24
15	Internal Leakage Data, Bendix T-slot Valve	25
16	Schematic Diagram of Hydraulic System	29
17	Servo Actuator/Mass Load Fixture Installation	30
18	Dynamic Response Test Instrumentation	32
19	Test System Gain Values	33
20	Pressure Level Switching Test Instrumentation	34
21	Schematic Diagram of Energy Consumption Test Instrumentation	36
22	View of Energy Consumption Test Instrumentation	37

LIST OF FIGURES (Cont'd)

Figure No.	Title	Page
23	Actuator Step Response Data, E-Systems Zero Lap Valve	40
24	Actuator Frequency Response Data, E-Systems Overlapped Valve	41
25	Actuator Pressure Level Switching Data, E-Systems Zero Lap Valve	44
26	Actuator Energy Consumption Data, E-Systems Overlapped Valve	47
27	Schematic Diagram of Pump Test Instrumentation	50
28	Pump Pressure Ripple Data	53
29	Pump Transient Response Data	55
30	Pump Performance Data	56
31	Schematic Diagram of LHS Simulator Hydraulic Systems	59/60
32	Test Actuator in LH UHT Load Module	61
33	UHT Actuator Load/Stroke Curve	61
34	FC-1 and FC-2 System Dual Pressure Pumps	62
35	Pressure Level Switching Data (actuator test), E-Systems Overlapped Valve	68
36	Pressure Level Switching Data (system test) E-Systems Zero Lap Valve	71
37	Pump Pressure Ripple Data, FC-1 System	74
38	Spectrum Analysis Data, FC-2 System	76
39	System Energy Consumption, E-Systems Zero Lap Valve	78

LIST OF TABLES

TABLE NO.	TITLE	PAGE
1	Transducer Information	18
2	Signal Conditioning and Readout Equipment	19
3	Quantitative Performance, Bendix Valves	26
4	Qualitative Performance, Bendix Valves	26
5	Quantitative Performance, E-Systems Valves	27
6	Qualitative Performance, E-Systems Valves	27
7	Actuator Test Parameters and Equations	38
8	Servo Actuator Dynamic Performance, Mass Load	42
9	Pressure Level Switching Data, Mass Load	45
10	Energy Consumption Data, Mass Load	48
11	Pump Test Parameters and Equations	51
12	Pressure Level Switching Data (Pump Test)	57
13	Servo Actuator Dynamic Performance, Force Load	66
14	Pressure Level Switching Data (Actuator Test)	69
15	Pressure Level Switching Data (System Test)	72
16	Spectrum Analysis Data	77
17	Energy Consumption Data, LHS Simulator	79

1.0 HARDWARE DEMONSTRATION PHASE (TASK II)

1.1 APPROACH

1.1.1 Introduction

The study phase, reported in Volume I, determined that certain energy conservation techniques can significantly reduce aircraft fuel consumption. Techniques considered to be effective in reducing losses and selected for laboratory demonstration testing were:

- o Control valves with overlap and shaped orifices
- o Dual pressure level systems

Specially designed hardware were fabricated to validate the potential of these energy saving techniques. Tests were then conducted to determine the energy consumption of the different techniques. In addition, dual pressure level switching transients were investigated, and actuator performance changes resulting from the use of the special control valves were examined. Section 1.1 describes the demonstration hardware, tests performed, and instrumentation used. Test procedures and results are presented in Section 1.2. A summary is given in Section 1.3.

1.1.2 Demonstration Hardware

1.1.2.1 Control Valves. Quiescent leakage in control valves causes appreciable power loss. Most of this internal leakage occurs at the spool null position and is a function of valve size, spool/sleeve clearance, and orifice geometry. Total internal leakage is the sum of the dynamic leakage that occurs when the valve is operating plus null leakage when the spool is stationary. Two approaches were investigated to determine potential energy savings: overlap and shaped orifices (see Section 2.4.5 in Volume I).

Valve design and performance requirements were established by a specification prepared to procure the valves. This specification is presented in Appendix A. Considerable latitude was permitted for deviating from the specification requirements providing the basic goals of the test program could be met with the proposed design. Cost was an important consideration in supplier selection.

Two companies were chosen to provide the test valves: Bendix Electrodynamics and E-Systems. Bendix provided both the valves and electronics on a consignment loan basis. E-Systems provided the valve electronics on a consignment basis. Descriptions of the test valves are given in the following paragraphs.

Bendix Electrodynamics. The test valve is a modified version of a 5,000 psi unit built for a prior Bendix project. The valve has separate motor and spool/sleeve assemblies that combine in one housing to form the valve assembly. The motor and spool both have rotary motion. Three interchangeable spool/sleeve assemblies provide three different configurations:

- o Zero overlap with linear slot orifices
- o High overlap with linear slot orifices
- o Small overlap with T-slot orifices

Spool travel is $\pm 10^{\circ}$ (± 0.0327 in.). Rated flow is 5 gpm. Multi-metering orifices are employed to minimize the maximum chip size the spool must shear. The maximum available shear-out force is 80 lb. Flow forces are exceptionally low due to special design features.

The motor is a brushless DC type with samarium-cobalt magnets to provide a high torque-to-power ratio, and is an "inside-out" design with magnets on the rotor and windings on the stator. The motor drives the spool directly; a torsional centering spring is attached to the opposite end of the spool. No seals are required on the spool. An RVDT provides the spool position feedback signal. The valve/motor assembly is shown on Figure 1.

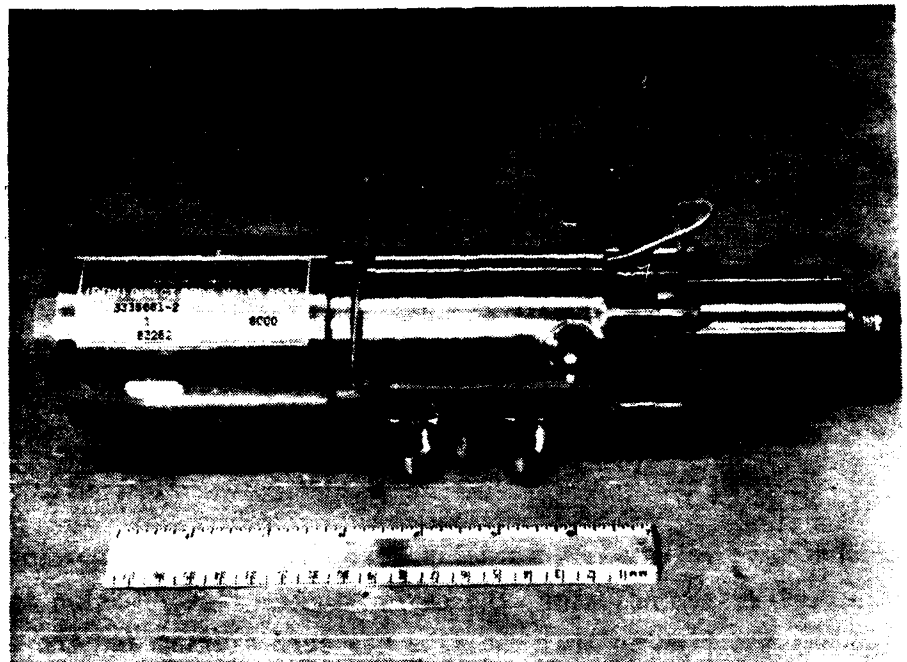


Figure 1. Bendix direct drive servo valve

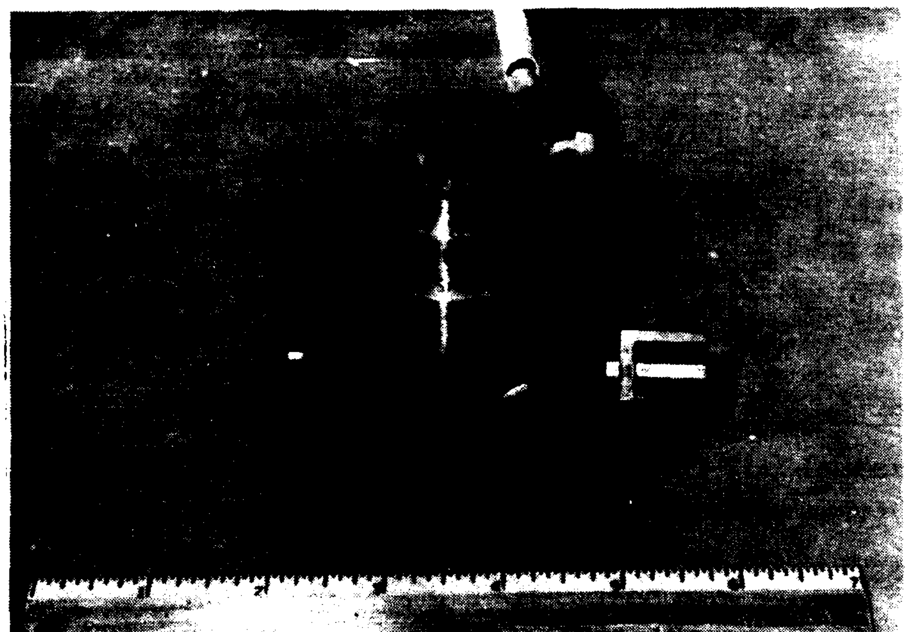


Figure 2. E-Systems direct drive servo valve

The Bendix electronics package drives the torque motor with an analog signal, and provides feedback loop closure around the valve spool and actuator piston. Loop gain adjustments are available.

NOTE: Flow gain, pressure gain, internal leakage, and frequency response tests were completed on the zero overlap and T-slot configurations (the high overlap valve had not yet been received). The T-slot valve failed during the servo actuator tests (see Section 1.2.2.2). No further tests were conducted on the Bendix valves due to program scheduling constraints.

E-Systems. The test valve was originally designed to control two independent systems on a dual system actuator, and has a single, linear-motion spool. The valve design was modified so that one side has zero overlap and the other side has 0.002 in. overlap. Rated flow is 3.5 gpm on each side. The valve mounting block is made so that only one system can be operated at a time. A linear motion LVDT is used to close the loop around the spool and provide a spool position signal. Available chip shear-out force is 45 pounds. The valve is shown on Figure 2.

The valve electronics package drives the force motor with a pulse-width-modulated signal, provides feedback loop closure around the valve spool, and has a manual bias control. A second electronics package provides loop closure around the servo actuator piston and has feedforward and feedback compensation adjustments.

1.1.2.2 Servo Actuator. The actuator was built by Vought for the LHS Advanced Development Program, reference 1, and was designed to operate the unit horizontal tail (UHT) on an A-7E test bed aircraft. The servo actuator has dual tandem cylinders and a dual tandem mechanical input control valve, Figure 3. Feedback is accomplished with mechanical linkages.

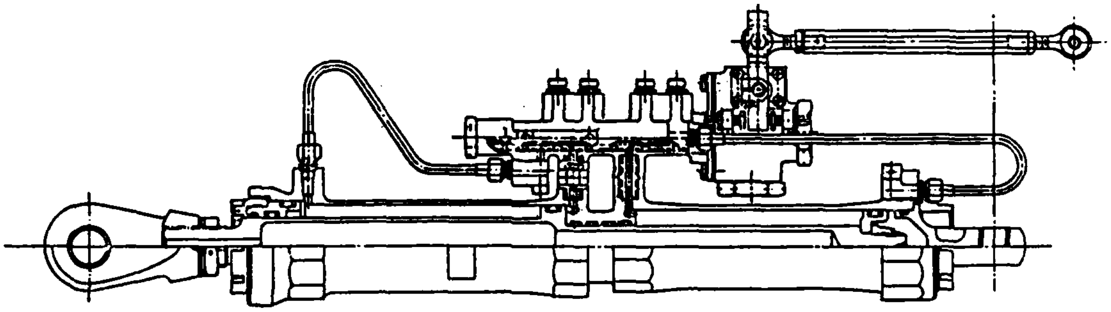


Figure 3. Original unit horizontal tail actuator

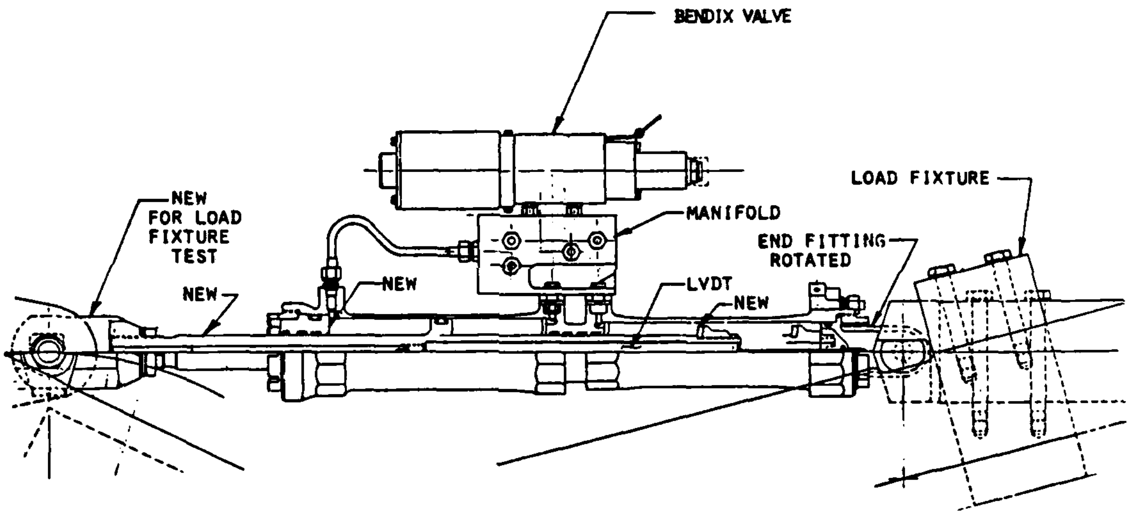


Figure 4. Modified UHT actuator with Bendix valve

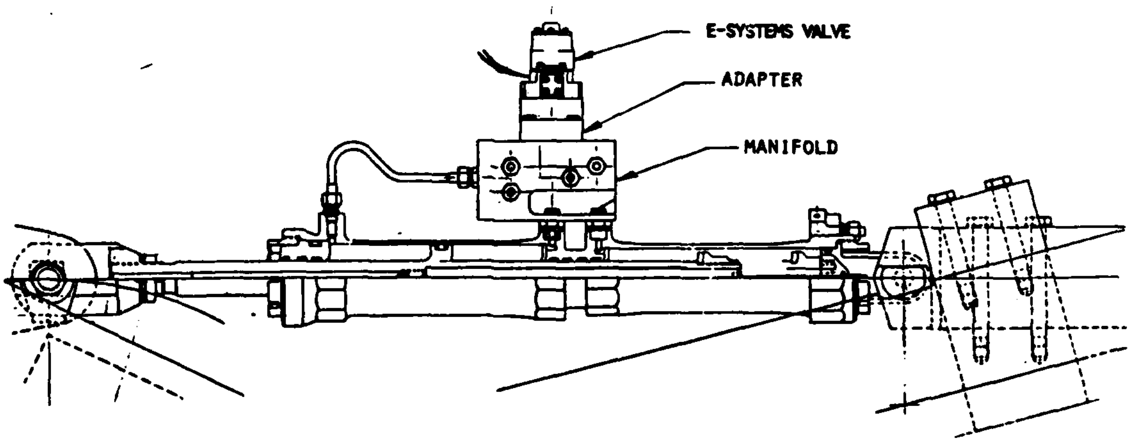


Figure 5. Modified UHT actuator with E-Systems valve

The UHT actuator was modified to mount it in a mass load test fixture and to accommodate the Bendix and E-Systems control valves. Changes made were:

- The FC-2 piston with unbalanced areas was replaced with a balanced area piston. The FC-1 side of the actuator was de-activated.
- A new piston rod seal cartridge was made to fit the new size piston rod.
- An LVDT was installed inside the piston rod to provide electrical feedback.
- A new rod end was fabricated to mate with the mass load fixture.
- The mechanical control valve housing was replaced with a manifold designed to interface with the Bendix and E-Systems test valves.

The modified UHT actuator with the test valve installations is shown on Figures 4 and 5. Major design parameters are:

Operating pressure:	8000 psi
Piston diameter:	2.368 in.
Rod diameter:	1.185 in.
Piston stroke:	6.58 in.
Extend/retract piston area	3.301 $in.^2$
Stall output force:	26,400 lb

1.1.2.3 Dual Pressure Pump. A 4000/8000 psi pressure level system offers substantial energy savings, reference Section 2.4.6 in Volume I. The dual pressure level concept employs a logic system to determine the pressure mode used. Important advantages obtained, in addition to energy savings, are reduced heat rejection and an increase in MTBF for hydraulic system components.

The test pumps were originally procured for the LHS Advanced Development Program, reference 1, and are a pressure compensatea, variable delivery axial piston design. The units, built by Vickers Aerospace Division, were designed for use on the LHS simulator, and were identified as M/N PV3-047-2, S/N 346580 and S/N 346581. Rated flow is 10 gpm at 5900 rpm. One pump (S/N 346581) has accumulated 1150 hours of endurance cycling during LHS programs; the other unit (S/N 346580) has accumulated 227 hours.

The pumps were modified to operate at two pressure levels by Vickers. The Statement of Work and Performance Requirements Specification Sheet are presented in Appendix A. The modification permits switching from a high pressure (8000 psi) to a low pressure (4000 psi) mode of operation and from the low pressure to the high pressure mode. Control pressure is ported to the pump compensator mechanism using a 3-way solenoid valve. Loss of electrical power to the valve will revert the pump to the high pressure mode. The modified pump, M/N PV3-047-4, is shown on Figure 6.

1.1.2.4 LHS Simulator. The original purpose for the simulator was to demonstrate the concept of using an 8000 psi operating pressure to achieve smaller and lighter weight hydraulic components than those used in aircraft with conventional 3000 psi systems, reference 1. The LHS simulator is a steel structure with hydraulic component installations designed to represent a full scale A-7E 8000 psi flight control system, Figure 7. A modular design approach was employed. Two types of modules are used:

Power Modules	Load Modules
FC-1 System	Aileron, LH
FC-2 System	Spoiler, LH
	UHT, LH & RH
	Rudder
	Speed Brake
	Leading Edge Flap
	LH Inboard, LH Outboard

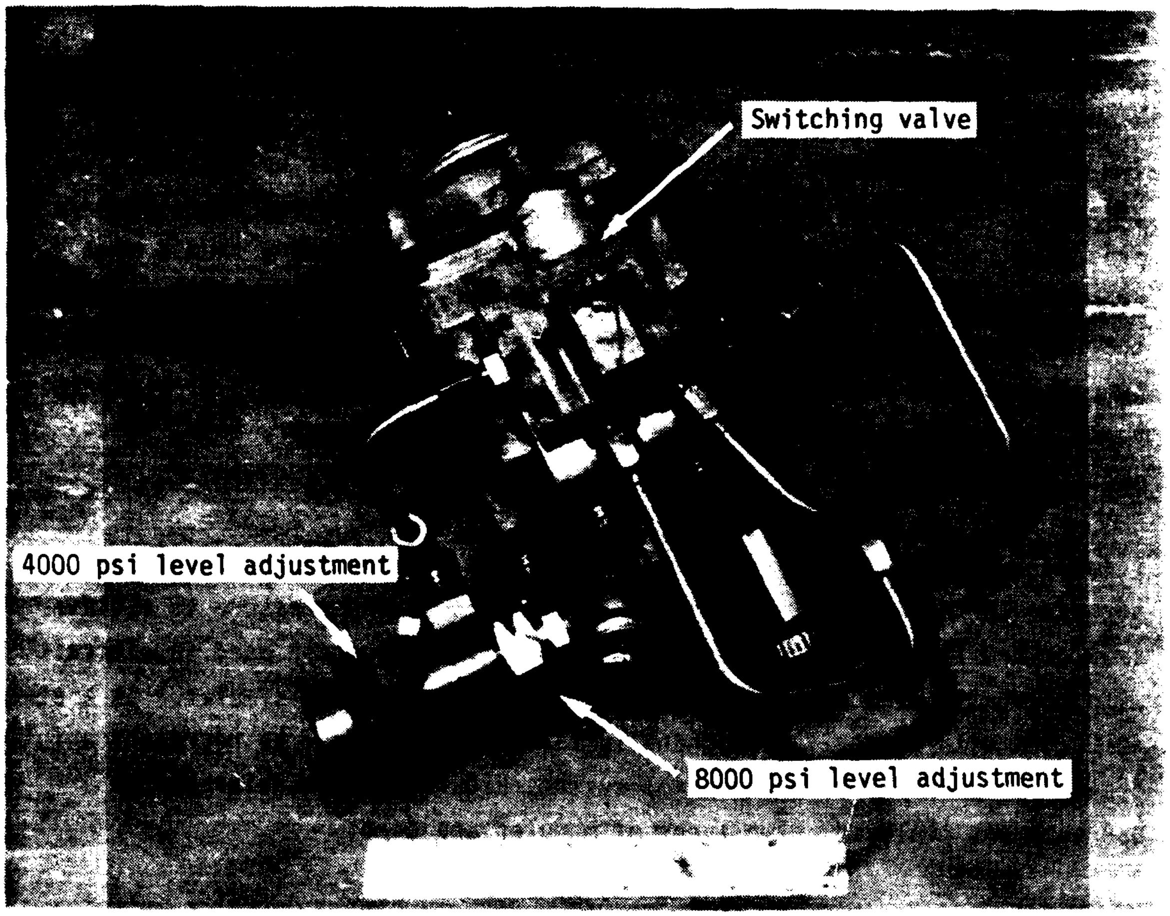


Figure 6. Vickers dual pressure pump with switching valve

Figure 7. Lightweight hydraulic system simulator

Each power module contains a pump, reservoir, filters, and valving to supply hydraulic fluid to the flight control actuators. Each actuator is mounted in a load module that duplicates the kinematics of an A-7E installation. Load/stroke conditions imposed on each actuator are based on specific, individual requirements. Twelve 8000 psi flight control actuators are installed in the simulator:

Primary Control Actuators	Secondary Control Actuators
Aileron, LH	Speed Brake
Spoiler, LH	Leading Edge Flap (4)
Roll Feel Isolation	
Unit Horizontal Tail, LH & RH	
Rudder	
AFCS (yaw)	

The simulator can be operated manually or automatically. Manual control is by "pilot stick" or through manipulating dials and switches on a console panel. Automatic control is by a mechanical programmer.

1.1.3 Tests Conducted

1.1.3.1 Control Valves. The primary objective was to determine the effects of overlap on valve flow gain, pressure gain, internal leakage, and frequency response. Flow gain is a plot of spool position versus no-load flow. Flow gain directly affects loop gain and therefore influences system stability. Pressure gain is a plot of spool position versus the difference in pressure between the valve cylinder ports which are temporarily blocked for this measurement. Pressure gain provides servo actuators with the capability to break away large friction loads with little error. Internal leakage is plotted as a curve of spool position versus return line flow with the cylinder ports blocked. Internal leakage dominates valve performance in the null region and is a continuous power loss. Frequency response involves plotting the output divided by the input (amplitude ratio in db) versus frequency. A valve with good response operates with minimum overshoot and undershoot and minimum phase lag within a required frequency band. Tests conducted were:

Test	Pressure Levels	Test Conducted By
Flow gain Pressure gain Internal leakage	4000 & 8000 psi	NAAO-Rockwell
Frequency response	3000 psi 8000 psi	Bendix E-Systems

1.1.3.2 Servo Actuator. The effects of valve overlap and dual pressure level operation on actuator performance were investigated by determining actuator response to step and sinusoidal inputs, by measuring piston position changes and pressure transients resulting from pressure level switching, and by comparing the energy consumption of the different valve/actuator combinations. The dynamic operation of servo actuators is typically characterized by the accuracy with which they track square wave commands and their ability to follow sinusoidal inputs over a given frequency band. Loop gains and command amplitude directly affect performance and must be carefully selected to provide valid test data. Switching pressure levels from 4000 to 8000 psi and 8000 to 4000 psi may produce undesirable side effects -- pressure transients and actuator position disturbances. The extent of these occurrences were to be determined. The energy consumption using an overlapped valve was compared to the energy consumption using a zero lapped valve. These measurements provided the basis for energy savings achieved.

Tests conducted on the servo actuator were:

Test	Actuator Configuration	Pressure Levels
Dynamic Response Pressure Level Switching (cycling flow) Energy Consumption	E-Systems Valve zero overlap 0.002 in. overlap	4000 & 8000 psi

1.1.3.3 <u>Dual Pressure Pump</u>. Pump performance was based on four operating characteristics: overall efficiency, pressure ripple, transient response, and heat rejection. Overall efficiency is delivered hydraulic power divided by input horsepower, and is measured at rated operating conditions of speed, flow, and temperature. Pressure ripple is caused by the stroking action of the pump pistons. This ripple causes standing pressure waves in the pump discharge line which can, if the right frequencies are present and amplitudes are sufficient, cause undesirable vibration in system tubing. Transient response is a measure of the ability of the pump to respond to rapid changes in flow demand. Slow response causes large amplitude pressure transients -- both overshoot and undershoot. Overshoot causes high stresses in system components; undershoot results in degraded actuator performance. Heat rejection is the result of internal leakage used to lubricate and cool the pump, and is caused by fluid throttling from high pressure to low pressure which raises fluid temperature. High fluid temperatures are harmful to seals and increases internal leakage rates in other components. Heat exchangers are frequently needed to remove heat added to systems by pumps with high heat rejection.

Tests conducted on the dual pressure pump were as follows:

Test	Pressure Levels
Overall Efficiency Pressure Ripple Transient Response Heat Rejection	4000 & 8000 psi
Pressure Level Switching (steady flows)	4000 & 8000 psi

1.1.3.4 LHS Simulator. The LHS simulator was used to determine the effects of overlapped valves and dual pressure levels on the operation of a full scale hydraulic system. The overall objective was to demonstrate that overlapped valves can be employed to conserve energy without seriously degrading actuator performance, and that switching operating pressure levels does not cause harmful pressure transients. The actuator was mounted in the LH UHT load module (see Figure 7). FC-1 system was used to power the actuator. FC-1 and FC-2 systems were operated for the pressure level switching tests.

Tests conducted on the LHS simulator were as follows:

Test	Configuration	Operating Mode	Pressure Levels
LH UHT ACTUATOR			
Dynamic Response	E-Systems Valve zero lap .002 in. overlap	step input sinusoidal input output loaded FC-1 only	4000 & 8000 psi
Pressure Level Switching	E-Systems Valve zero lap .002 in. overlap	output motionless output moving output loaded FC-1 only	4000 & 8000 psi
Energy Consumption	E-Systems Valve zero lap .002 in. overlap	sinusoidal output output unloaded output loaded FC-1 only	4000 & 8000 psi
SYSTEM			
Pressure Level Switching	FC-1 & FC-2 systems	All actuators cycling Operating modes: 2%, 10%, and 50% load/stroke	4000 & 8000 psi

Test	Configuration	Operating Mode	Pressure Levels
Pressure Ripple	FC-1 & FC-2 systems	All actuators at null	4000 & 8000 psi
Spectrum Analysis	FC-1 & FC-2 systems	All actuators at null Pump speed sweep	4000 & 8000 psi
Energy Consumption	E-Systems Valve zero lap .002 in. overlap	LH UHT actuator cycling All other actuators at null LH UHT unloaded LH UHT loaded FC-1 only	4000 & 8000 psi

1.1.4 Instrumentation

1.1.4.1 Hydraulic Test Bench. The performance of the Bendix and E-Systems valves was evaluated using a bench designed for testing 8000 psi hydraulic components, Figure 8. The test stand, built by Dayton T. Brown, is capable of delivering flows up to 18 gpm at pressures up to 8000 psi. Flow is measured by a positive displacement meter with readout in any desired units on a microprocessor based indicator. Fluid temperature can be controlled at levels between +100 and +200^{0}F. The bench contains fluid per specification MIL-H-83282; fluid filtration is 3 microns absolute.

1.1.4.2 Electronic Data Analysis System. A multi-channel computer based data analysis system was used to support all tests, Figure 9. A block diagram of the system and component identification are given on Figure 10. Signal processing is performed on an analyzer with pre-programmed waveform analysis/manipulation functions for both time and frequency domain measurements. Capabilities include transient analysis, spectrum analysis, and mathematical operations from basic to complex such as fast fourier transforms. The analyzer is also a digital oscilloscope with flexible multi-trace display capabilities. Bandwidth is 100 KHz with 14 bit A/D resolution; accuracy is 0.1%.

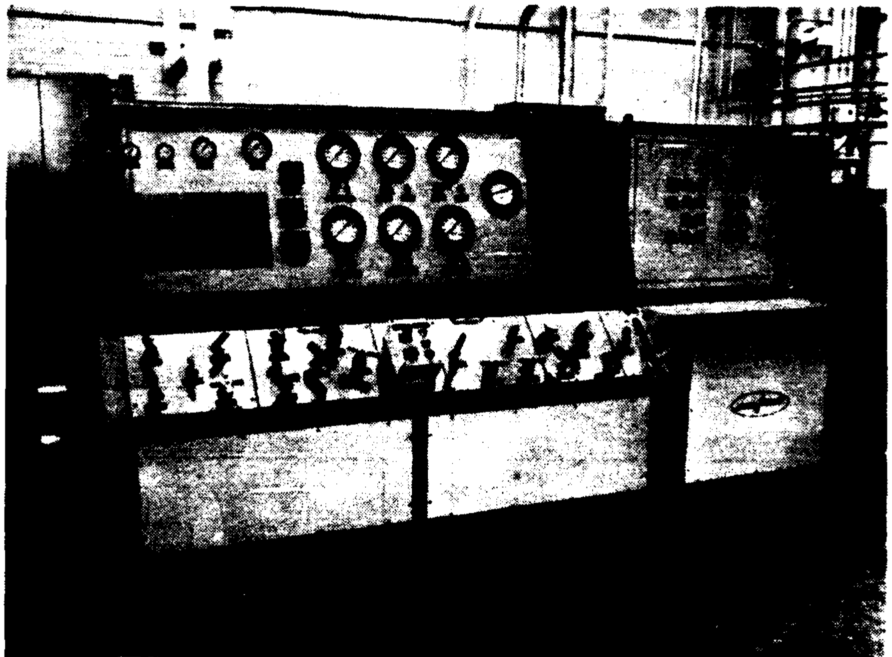


Figure 8. 8000 psi hydraulic test bench

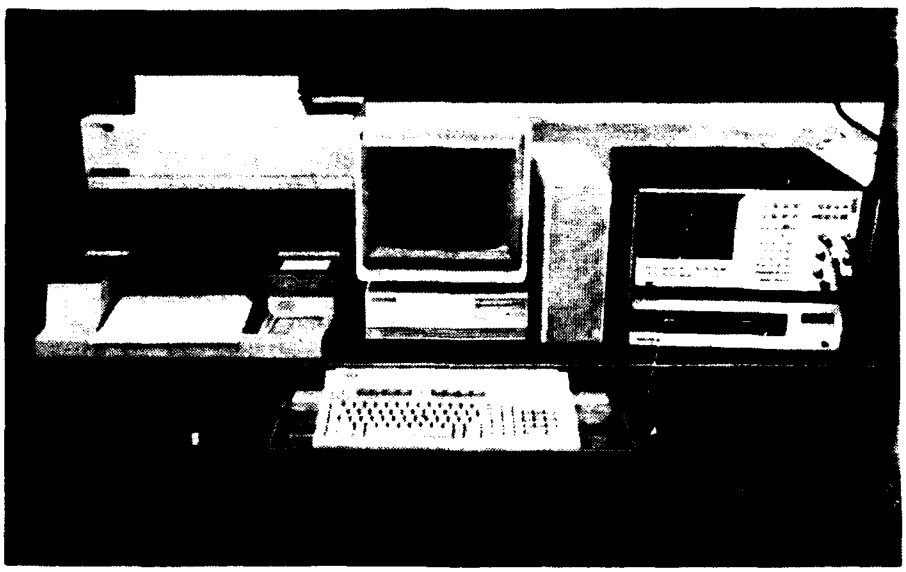
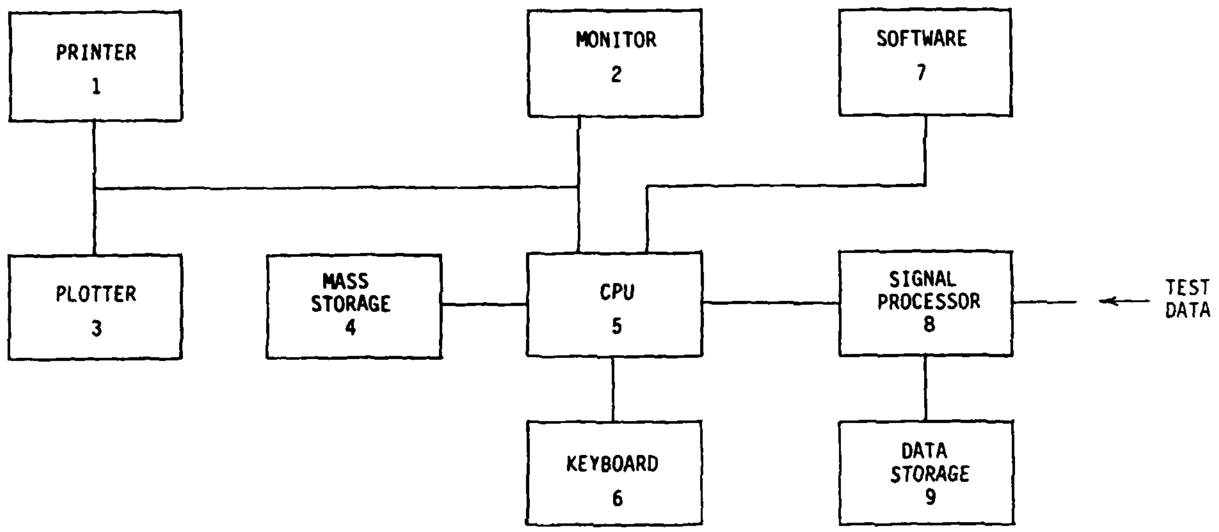


Figure 9. Electronic data analysis system



COMPONENT	PART NUMBER	MANUFACTURER
1	2932A	Hewlett-Packard
2	35741A	" "
3	7475A	" "
4	9153A	" "
5	9000 Series 300	" "
6	46021A	" "
7	EASY	Entek
8	6100	Data Precision
9	681	" "

Figure 10. Block diagram of data analysis system

Operation of the data processor is enhanced through the use of a CPU and customized software. Soft keys located on the computer keyboard are pre-programmed to execute special functions. Menu driven programs were employed to make various types of graphs on an inter-active X-Y plotter. Special data presentation capabilities include order tracking and three-dimensional cascade displays.

1.1.4.3 Transducers. Several different types of transducers were employed in the demonstration tests to measure pressure, flow, temperature, torque, and speed; these are listed on Table 1. The transducers produce electrical signals which must be conditioned; signal conditioning and readout equipment are listed in Table 2.

All transducers were calibrated so that calibration factors could be developed for use in the data analysis system. The system has provisions for entering engineering units per volt, such as in^3/sec/volt, psi/volt, lb-in/volt, and rpm/volt.

1.2 HARDWARE DEMONSTRATIONS

1.2.1 Control Valves

1.2.1.1 Procedure. Schematic diagrams of the setups used for the flow gain, pressure gain, and internal leakage tests are presented in Figure 11. Command signal waveforms and frequencies are also given on Figure 11. The test data are based on spool displacement, therefore different displacement amplitudes were employed for the Bendix and E-Systems valves. These amplitudes, given on Figure 11, were based on the signal amplitude generated by the spool position feedback transducer (LVDT or RVDT). Spool displacements used in the flow gain and internal leakage tests were the same. Spool displacement used to determine pressure gain was approximately 25% of that used in the other tests. All tests were conducted with valve inlet pressures of 4000 and 8000 psi; return pressure was approximately 100 psi. Test bench fluid temperature was controlled at 120 $\pm 5^{\circ}$F. A view of the flow gain test setup with the E-Systems valve is shown on Figure 12.

TABLE 1. Transducer information

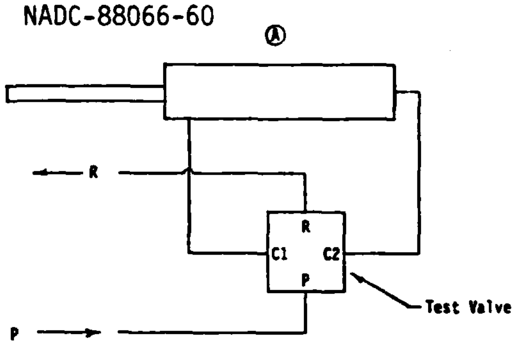
TEST	PARAMETER	TYPE	MODEL NO.	MANUFACTURER	REMARKS
Control Valves Servo Actuator Dual Press. Pump	Pressure	Strain Gage	122EF76	Viatran	20 KHz bandwidth
Dual Press. Pump LHS Simulator	Pressure	Piezoelectric	108A02	PCB Piezotronics	150 KHz bandwidth
Dual Press. Pump LHS Simulator	Flow	Turbine	10C1510A	Fischer & Porter	Modulated frequency signal conditioning
Control Valves Servo Actuator LHS Simulator	Flow	Metering Cylinder	514C-1	Industrial Measurements	Integral velocity transducer Relief valves in piston
All	Temperature	Thermocouple	None	Rockwell International	Chromel-alumel Copper-constantan
Servo Actuator Dual Press. Pump LHS Simulator	Torque	Stain Gage	1615K123	Lebow Products	Signal transmission by rotary transformer
Servo actuator Dual Press. Pump LHS Simulator	Speed	Magnetic Pickup	3010-AN	Electro Corp.	Mounted near: 60 tooth gear 128 tooth gear

TABLE 2. Signal conditioning and readout equipment

	SIGNAL CONDITIONING			READOUT		
TRANSDUCER	TYPE	M/N	MANUFACTURER	TYPE	M/N	MANUFACTURER
Pressure (strain gage)	Bridge balance & DC amplifier	9608	Tegam		*	
Pressure (piezoelectric)	Power supply	484B	PCB Piezotronics		*	
Turbine meter	Oscillator/ pre-amplifier	55GE2238	Fischer & Porter	Microprocessor w/digital readout	ITK 7650	NES, Inc.
	Frequency to DC converter	SD103	Spectral Dynamics			
Actuator LVDT	Demodulator DC amplifier	ATA-101 5503	Schaevitz Bay Laboratories	Digital Multimeter	3466A *	Hewlett-Packard
Flow cylinder	DC amplifier	5503	Bay Laboratories		*	
Temperature				Microprocessor w/digital readout	205	Doric
Torque (rotary transformer)				Digital indicator	7540	Lebow
Speed	Frequency to DC Converter	7540	Lebow	Digital indicator	5512A	Hewlett-Packard

* Data processing system (see Section 1.1.4.2)

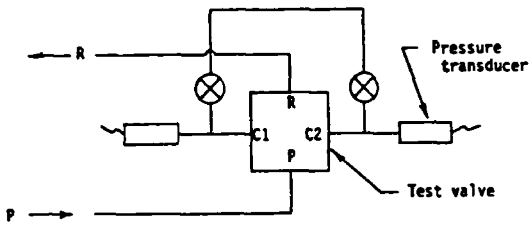
Flow
Gain
Setup



Command signal wave form:	Sine (single cycle)
Wave period:	10 sec
Feedback signal amplitude:	
Bendix valve	±3.5 V (F.S.)
E-Systems valve	±10.5 V (F.S.)

NOTE: Flow rate limited to 3 gpm due to capacity of flow metering cylinder Ⓐ.

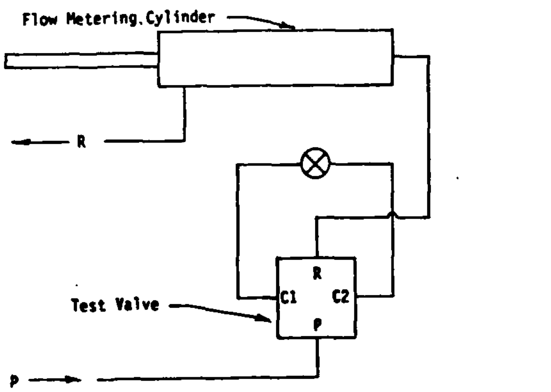
Pressure
Gain
Setup



Command signal wave form:	Ramp (single cycle, p-p)
Wave period:	100 sec
Feedback signal amplitude:	
Bendix valve	±0.9 V (F.S.)
E-Systems valve	±2.0 V (F.S.)

NOTE: Fluid volume at C1 and C2 was as small as practical.

Internal
Leakage
Setup



Command signal wave form:	Cosine (single cycle)
Wave period:	100 sec
Feedback signal amplitude:	
Bendix valve	±3.5 V (F.S.)
E-Systems valve	±10.5 V (F.S.)

NOTE: Fluid volume at C1 and C2 was equal and approximately 0.6 in^3 at each port.

Figure 11. Schematic diagrams of valve test setups

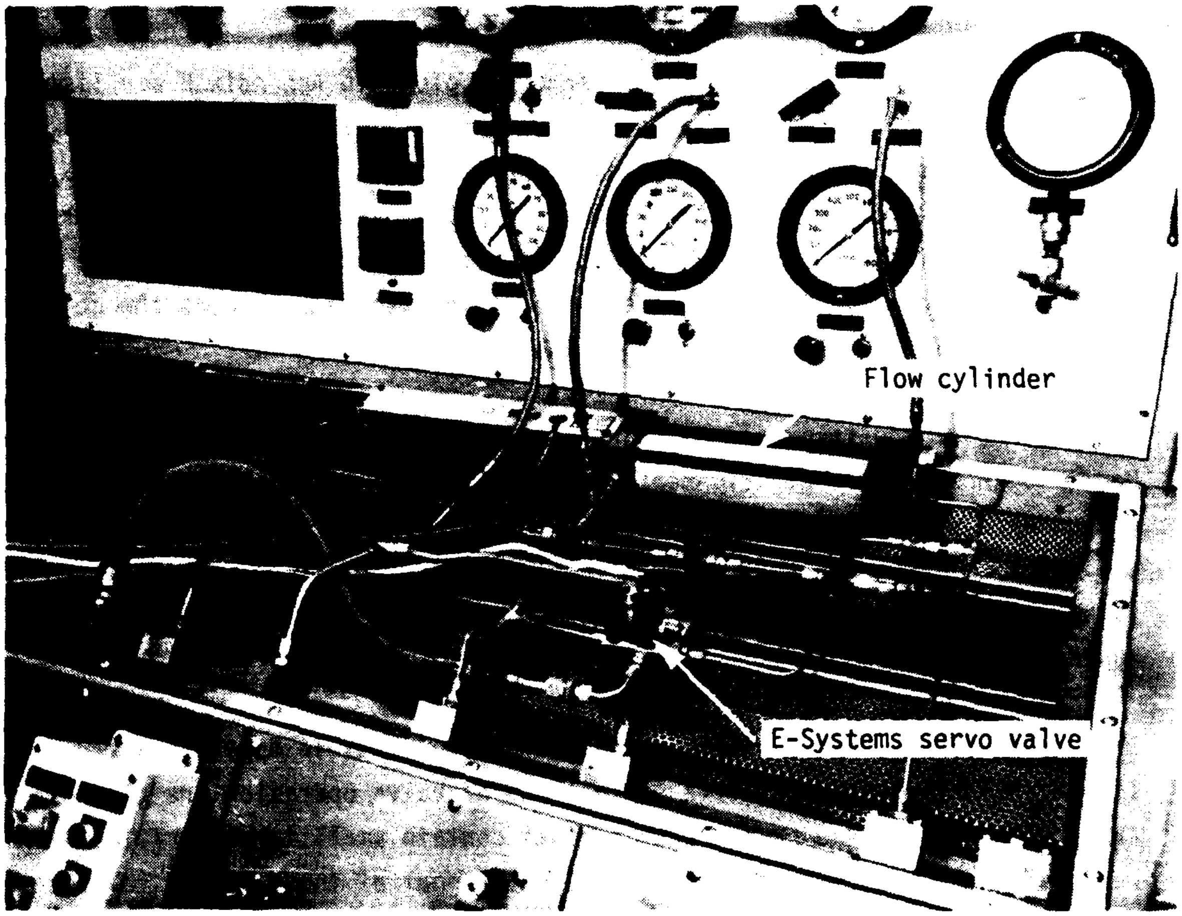


Figure 12. View of flow gain test setup

1.2.1.2 Results. Flow gain, pressure gain, and internal leakage plots made by the data analysis system are presented in Appendix B. Examples of this data are shown on Figures 13, 14, and 15. Results of the Bendix and E-Systems valve tests are discussed in the following paragraphs.

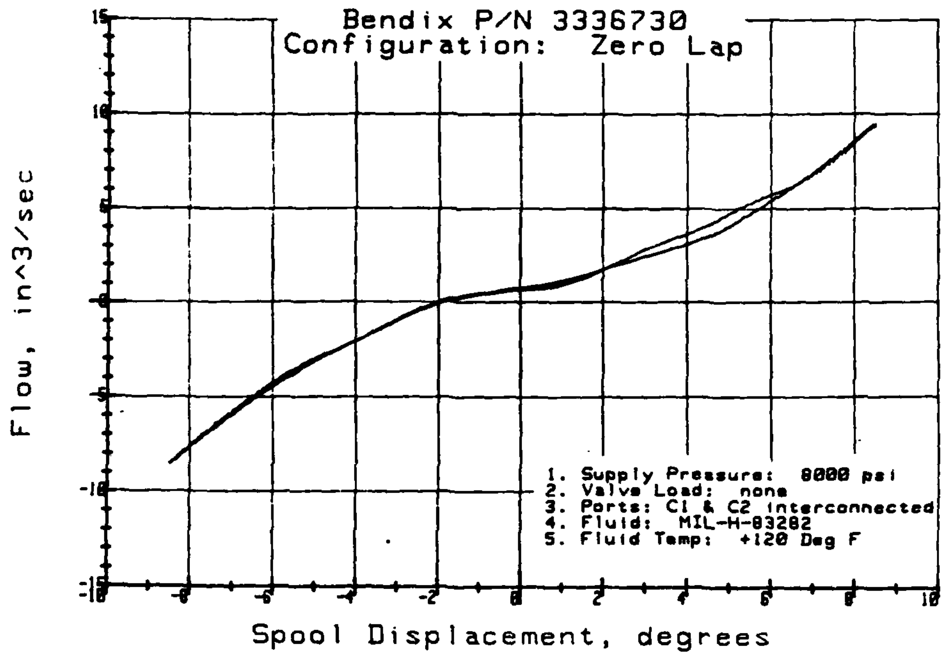
Bendix. Performance values obtained from the plots in Appendix B are listed in Table 3. Qualitative evaluations of valve operation are given in Table 4. The flow and pressure gain curves have a null offset caused by the Bendix electronic package. Symmetrical signals applied to the input of the package were unsymmetrical at its output. Bias adjustments are available inside the package; none are available outside. No attempt was made to bias the command input signal in order to balance the output.

Performance of the zero lap configuration was affected by internal friction. Bendix acknowledged that alignment between the spool axis and torque motor output axis were not as good as desired because of adverse tolerance build-ups. It should be noted that the T-slot configuration was installed in valve assembly P/N 3335661-2; the zero lap spool was installed in valve assembly P/N 3336730.

E-Systems. Performance values obtained from the plots in Appendix B are listed in Table 5. Qualitative evaluations of valve operation are given in Table 6. The flow and pressure gain plots contain small irregularities. This was due to the pulse-width-modulation (PWM) signal of the E-Systems electronic package. The PWM in effect imposed a small dither motion on the spool, and was most evident on the pressure gain plots. The horizontal irregularities are due to small spool displacements sensed by the LVDT; the vertical irregularities are due to pressure fluctuations caused by the small spool displacements. Although the plots do not have a neat appearance, the "noise" was not considered detrimental to valve performance.

PWM caused problems during the entire test program because of the "electrical noise" it generates. Considerable effort was required to shield other test instrumentation from conducted and radiated PWM noise so that noise-free data could be obtained. PWM caused the E-Systems valve to work hard, and on some occasions the motion of the servo actuator piston was observed to contain high frequency, low amplitude irregularities.

FLOW GAIN



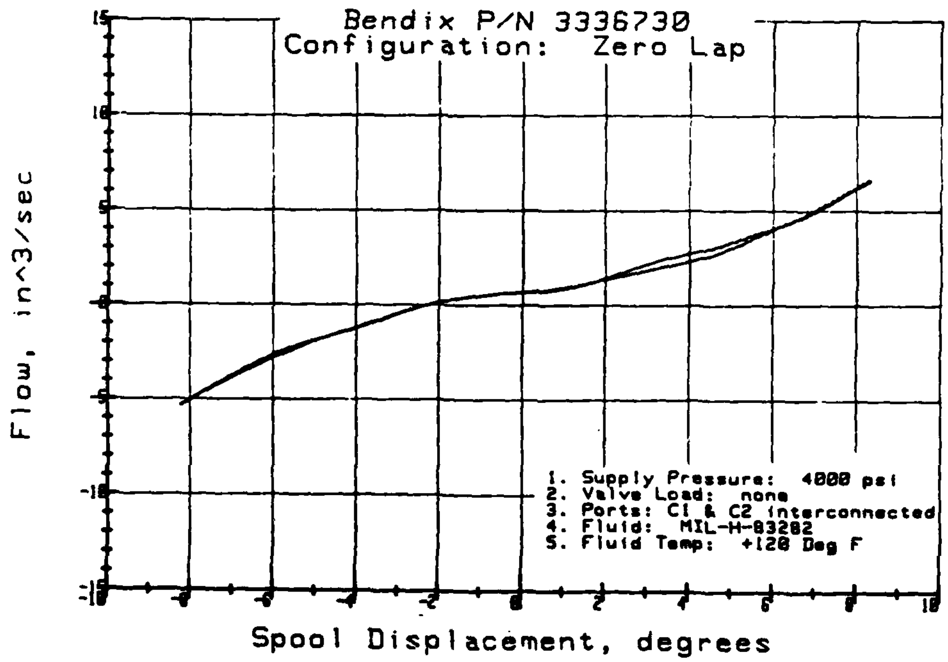
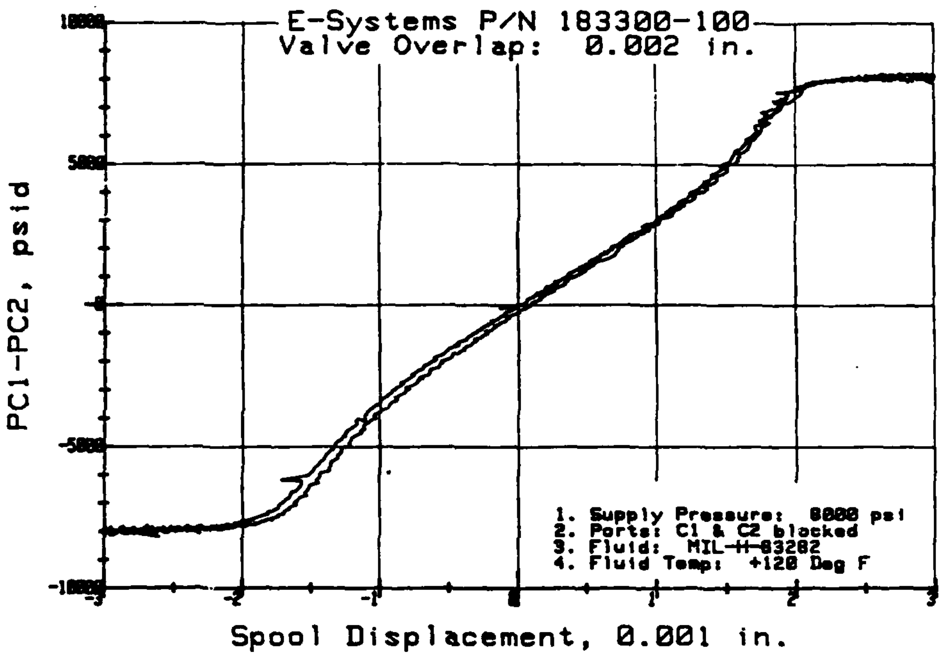


Figure 13. Flow gain data, Bendix zero lap valve

PRESSURE GAIN



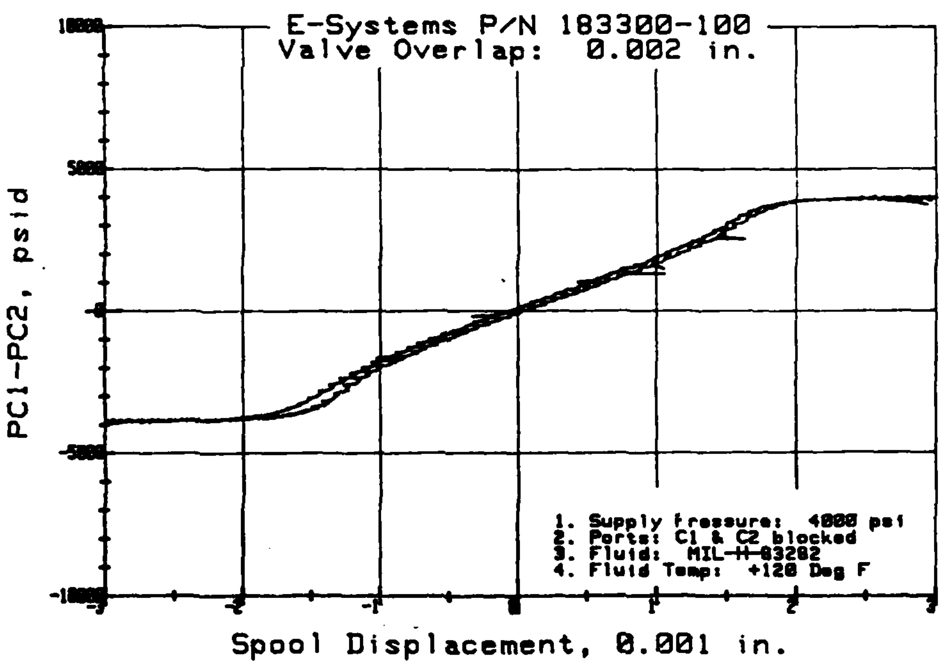
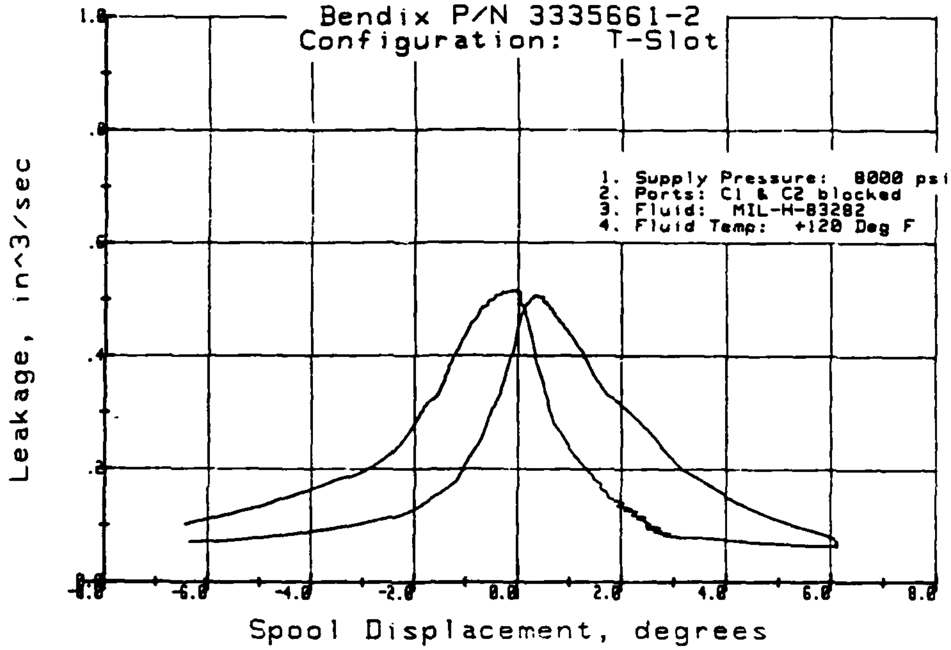


Figure 14. Pressure gain data, E-Systems overlapped valve

INTERNAL LEAKAGE



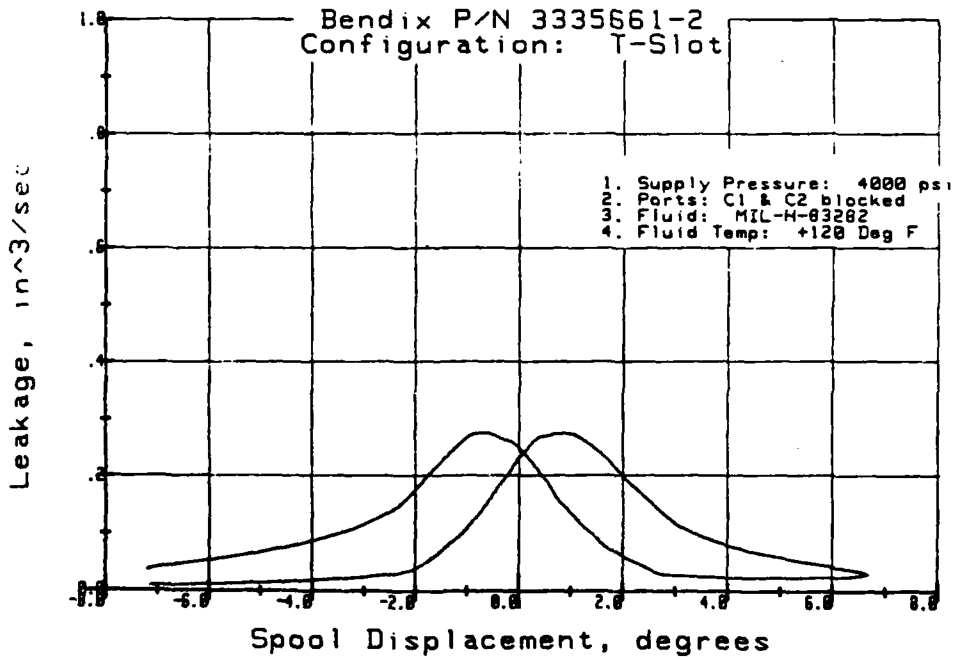


Figure 15. Internal leakage data, Bendix T-slot valve

TABLE 3. Quantitative performance, Bendix valves

	3rd QUADRANT	NULL	1st QUADRANT
FLOW GAIN, in^3/sec/deg			
Zero Lap Valve			
4000 psi	0.71		0.64
8000 psi	1.06		1.00
T-Slot Valve			
4000 psi	1.63		1.27
8000 psi	2.57		1.88
PRESSURE GAIN, psi/deg			
Zero Lap Valve			
4000 psi	8511		1828
8000 psi	13,072		4846
T-Slot Valve			
4000 psi	4250		4250
8000 psi	8016		8016
INTERNAL LEAKAGE, in^3/sec/deg			
Zero Lap Valve			
4000 psi		.057	
8000 psi		.105	
T-Slot Valve			
4000 psi		.275	
8000 psi		.512	

TABLE 4. Qualitative performance, Bendix valves

	GAIN BALANCE	GAIN LINEARITY	HYSTERESIS	NULL AREA GAIN CHANGE
FLOW GAIN				
Zero Lap	good	fair	good	high
T-Slot	fair	excellent	good	high
PRESSURE GAIN				
Zero Lap	poor	poor	good	high
T-Slot	excellent	excellent	excellent	none

TABLE 5. Quantitative performance, E-Systems valves

	3rd QUADRANT	NULL	1st QUADRANT
FLOW GAIN, $in^3/sec/in$			
Zero Lap Valve			
4000 psi	895		895
8000 psi	1351		1351
.002 Overlap Valve			
4000 psi	918		896
8000 psi	1339		1336
PRESSURE GAIN, psi/in			
Zero Lap Valve			
4000 psi	13.4×10^6		13.4×10^6
8000 psi	21×10^6		21×10^6
.002 Overlap Valve			
4000 psi	2.94×10^6	1.82×10^6	2.78×10^6
8000 psi	5.46×10^6	3.09×10^6	5.80×10^6
INTERNAL LEAKAGE, $in^3/sec/in$			
Zero Lap Valve			
4000 psi		0.29	
8000 psi		0.39	
.002 Overlap Valve			
4000 psi		0.023	
8000 psi		0.067	

TABLE 6. Qualitative performance, E-Systems valves

	GAIN BALANCE	GAIN LINEARITY	HYSTERESIS	NULL AREA GAIN CHANGE
FLOW GAIN				
Zero Lap	excellent	excellent	good	none
.002 Overlap	excellent	excellent	excellent	high
PRESSURE GAIN				
Zero Lap	excellent	excellent	excellent	none
.002 Overlap	excellent	excellent	excellent	high

Frequency Response. Frequency response tests were conducted by Bendix and E-Systems; the plots are presented in Appendix B. Bendix ran their tests at 3000 psi using MIL-H-83282 fluid, E-Systems conducted a single test at 8000 psi using CTFE fluid. A summary of results is given below:

	Spool Displacement	-3 db Point Frequency, Hz	-3 db Point Phase angle, deg
Bendix Valves			
Zero Lap	±10%	50	-57
	±25%	53	-59
T-Slot	±10%	77	-123
	±25%	81	-106
E-Systems			
.002 in. Overlap	±25%	130	-43

1.2.2 Servo Actuator

1.2.2.1 Procedure. Three different types of tests were conducted: dynamic response, pressure level switching, and energy consumption. Each test employed different instrumentation and methods of acquiring the data. A general setup was used, however, for all three tests. A schematic diagram of the hydraulic system is presented on Figure 16.

The servo actuator was mounted in a fixture designed to simulate the mass load of a horizontal stabilizer on the RA-5C airplane, Figure 17. The fixture was originally built to test an actuator developed for the LHS program, reference 2. Installation of the UHT actuator in the fixture required new mounting provisions; these are indicated in Figure 17. The fixture was made very rigid to minimize structural dynamics. The effective mass load on the actuator was 6.94 lb-sec^2/in. Prior tests conducted on the fixture have established that a mass load/actuator natural frequency occurs in the range of 19 to 23 Hz

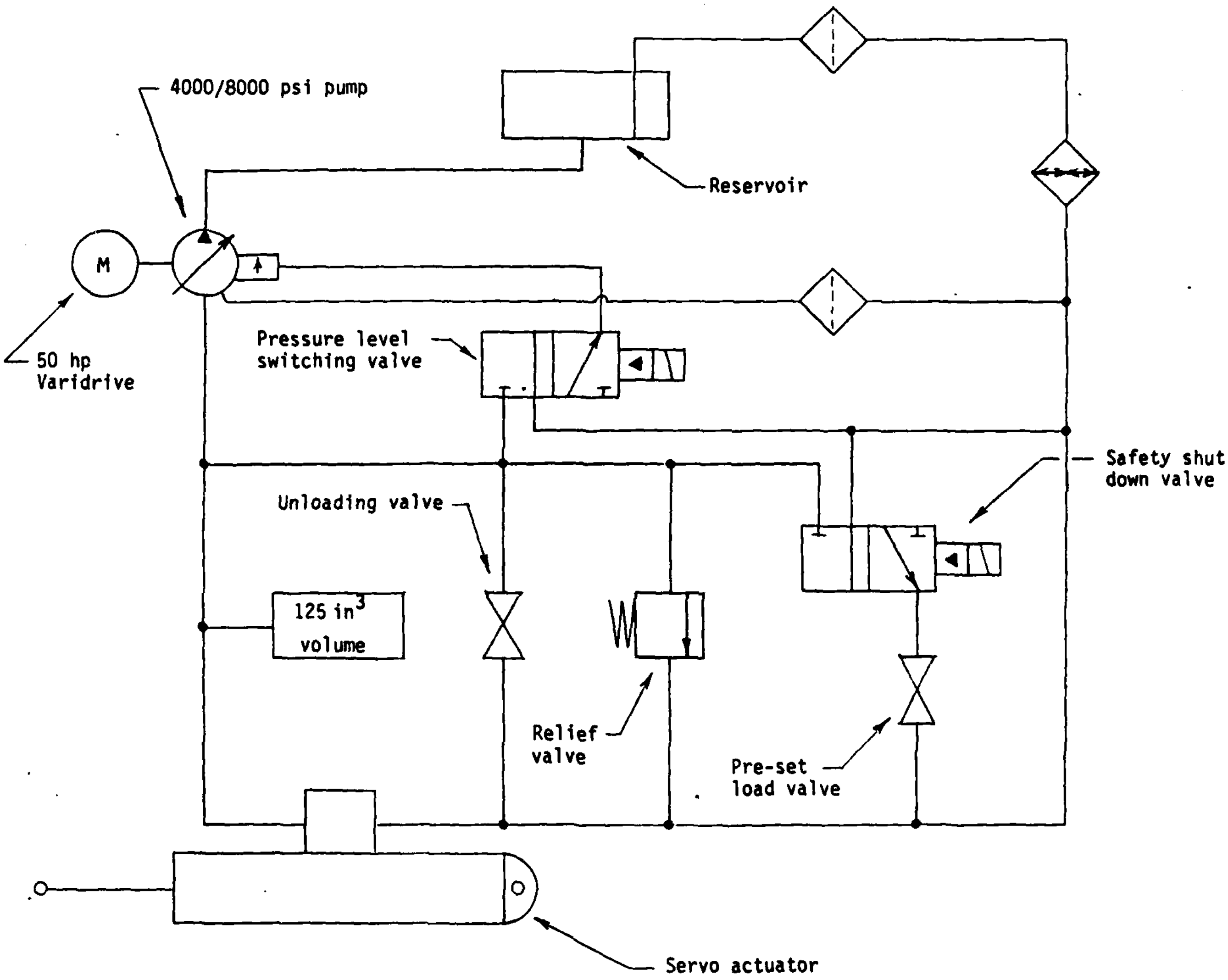


Figure 16. Schematic diagram of hydraulic system

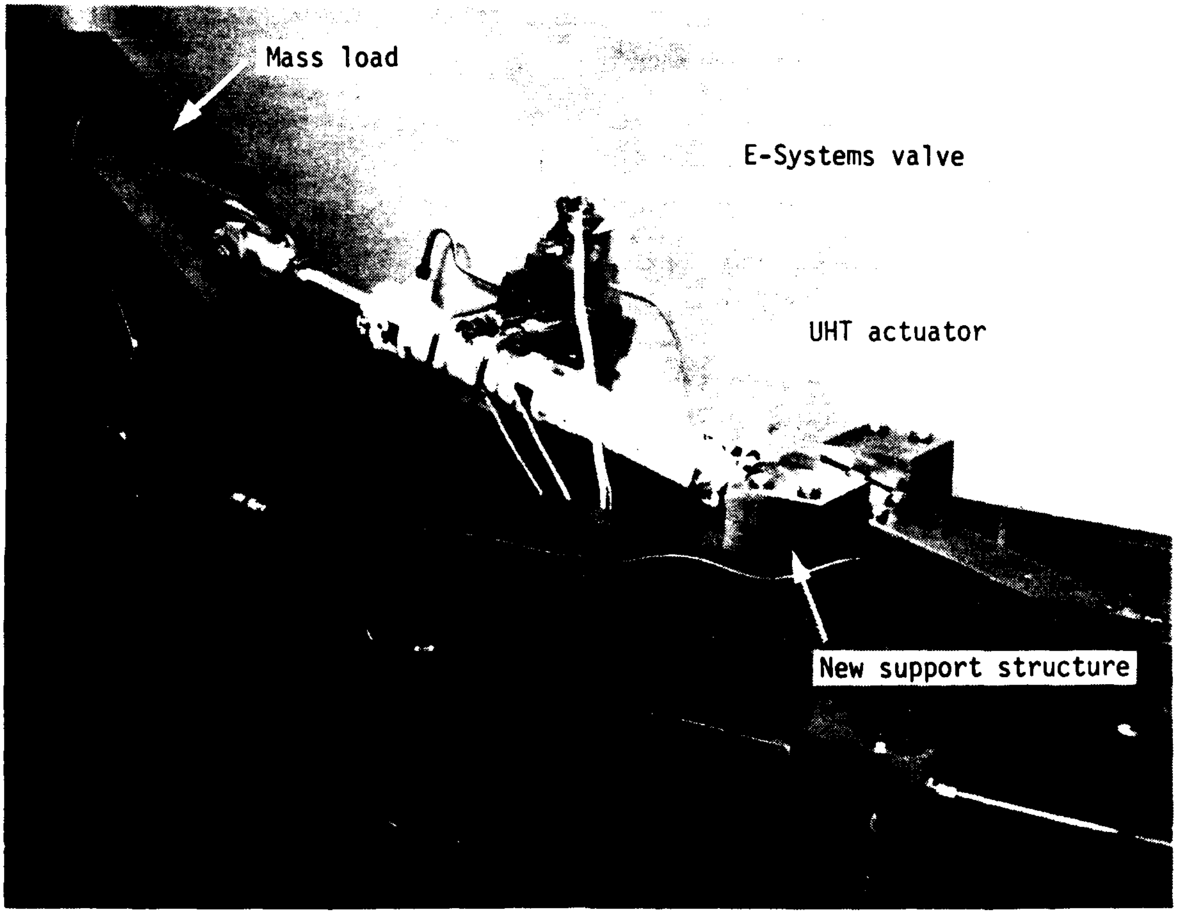


Figure 17. Servo actuator/mass load fixture installation

(depending upon the actuator and fluid being used). Operation in this frequency region could cause physical damage to test parts. An upper limit of 18 Hz was therefore used for frequency response tests conducted with the mass load.

Dynamic Response Tests. A schematic diagram of the instrumentation is shown on Figure 18. The function generator signal was used as the valve input signal in the frequency response tests because: 1) the E-Systems electronic drive unit employed PWM; and 2) the response characteristics of the EDU would have a negligible effect on test results. The feedback and loop gains were optimized by observing actuator performance with step inputs and using settings that produced slight ringing with operation at 8000 psi. These selected gain values were maintained during all tests. A block diagram showing gain values is shown on Figure 19.

The step input tests were conducted using an actuator output of 0.100 in. p-p at 1 Hz. The frequency response test was performed using an input that produced an output of 0.100 in. p-p at 0.4 Hz and maintaining this input constant over the frequency sweep range. The step input and frequency response tests were conducted with and without the mass load on the actuator.

Pressure Level Switching. The effect of switching operating pressure level on actuator piston position was determined with the piston both motionless and moving during the pressure level switchover. A block diagram of the instrumentation is shown on Figure 20. All tests were conducted with the actuator piston attached to the load mass. Piston motion was sinusoidal with 0.250 in. peak-to-peak travel at 1 Hz. Data were acquired with the pressure level switchover made at 0^{o} and at 90^{o} during the sinusoidal motion tests. Loop gain values were the same as those used for the dynamic response tests (see Figure 19).

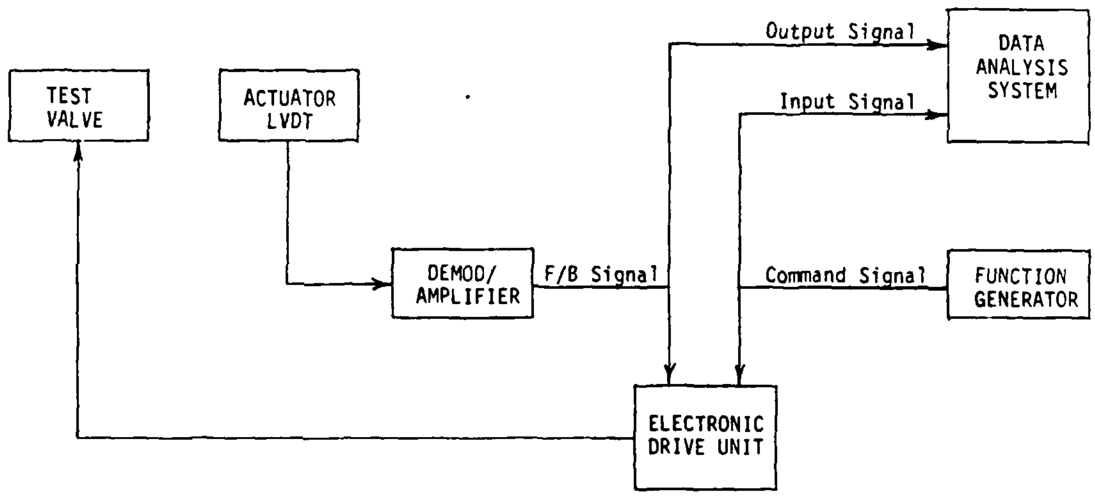


Figure 18. Schematic diagram of dynamic response test instrumentation

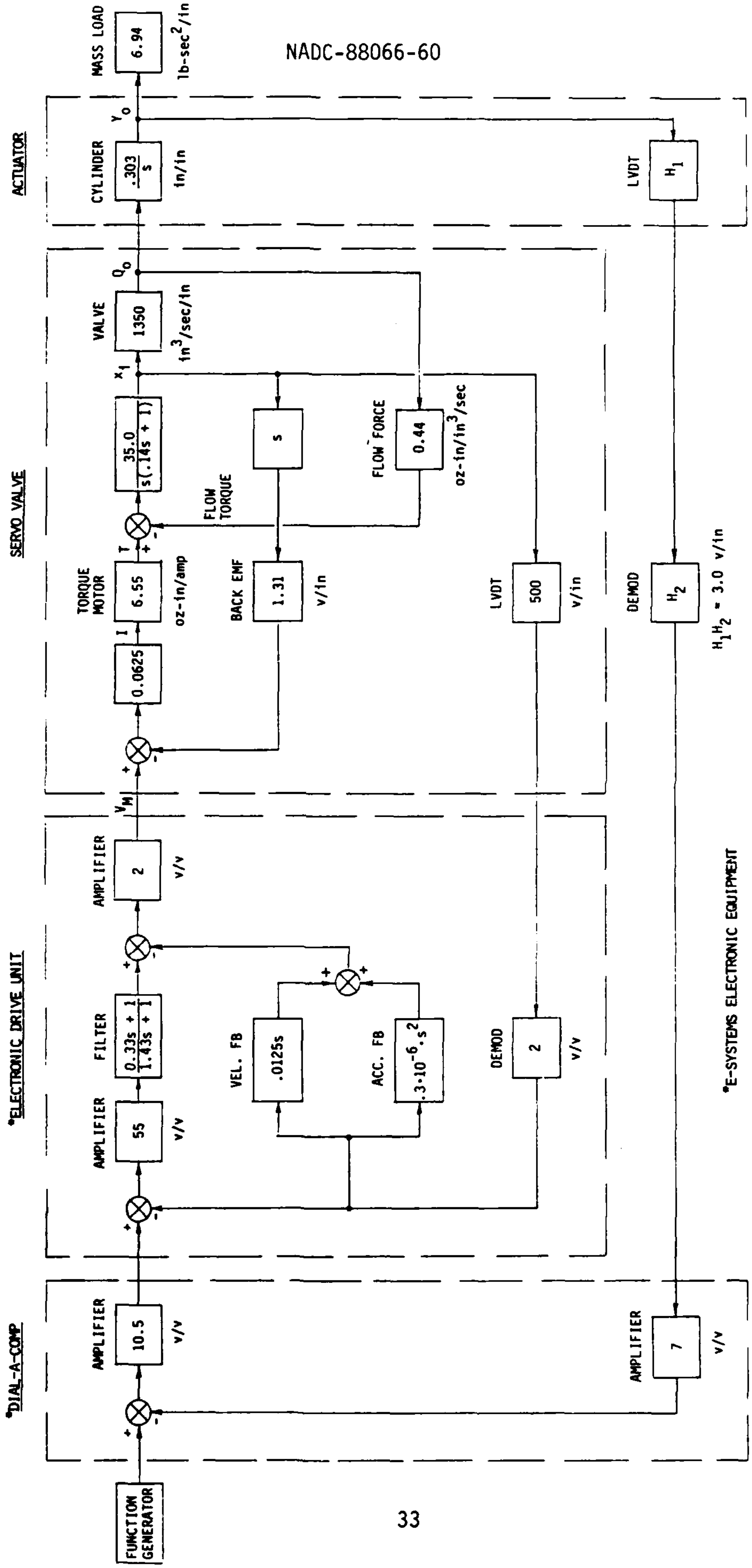


Figure 19. Test system gain values

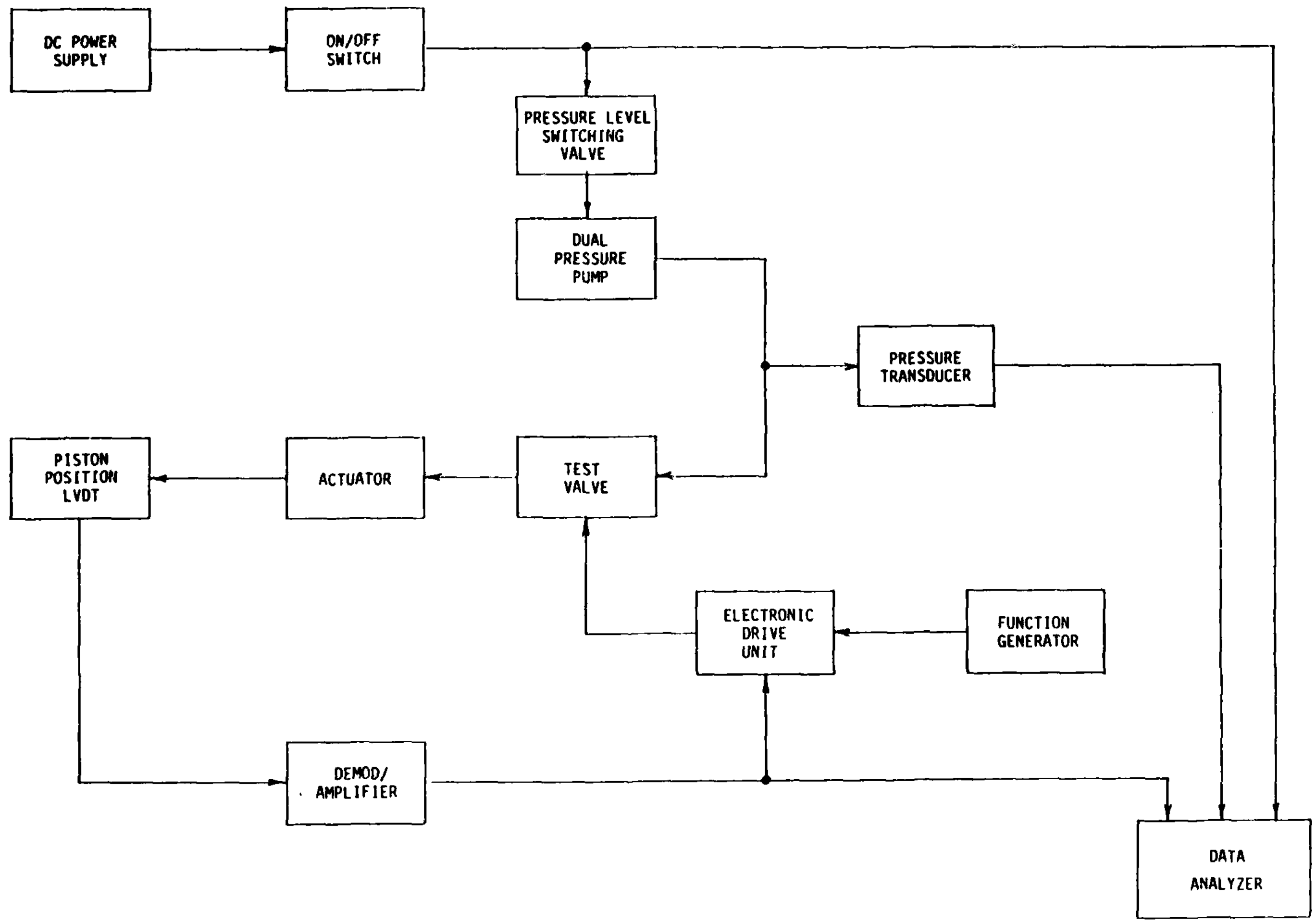


Figure 20. Pressure level switching test instrumentation

Energy Consumption Tests. A schematic diagram and photograph of the instrumentation are shown on Figures 21 and 22, respectively. The heat rejection of the pump, servo actuator, and system were determined while the actuator was operated sinusoidally over a frequency range of 1 to 4 Hz. The data were taken at the point of maximum power consumption, i.e., at the 45^{o} point on the output wave form.[1] This was done by using the "clock" input on the analyzer, and applying a sinusoidal signal to this input that was 45^{o} out-of-phase with the actuator output. The clocking signal triggered the analyzer to take data at each zero crossing having a positive slope (clocking signal). The time duration of the sweep from 1 Hz to 4 Hz was limited by the piston travel in the flow cylinder and was approximately 25 seconds. The input amplitude was held constant over the frequency sweep range, and was a value that produced an actuator output of 0.100 in. p-p at 0.4 Hz. Testing was conducted with and without the mass load on the actuator. The pump, actuator, and system heat rejections were calculated by the data analyzer using the parameters and equations given in Table 7. Pump speed was 5400 rpm. It should be noted that pump S/N 422717 was used for these tests. (Pumps S/N 346580 and S/N 346581 had not yet been received, see Section 1.1.2.3.)

[1] Flow and pressure in the actuator were sinusoidal in form and 90^{o} out-of-phase. Power consumed by the actuator is: Power = $(\Delta P \cdot \sin\theta \cdot Q \cdot \cos\theta) \div 1714$, where ΔP = PC1 - PC2 in psid and Q = flow in gpm. Maximum power occurs when $\theta = 45^{o}$ and is: Power = $0.5 \cdot \Delta P \cdot Q \div 1714$. Maximum actuator efficiency also occurs when $\theta = 45^{o}$.

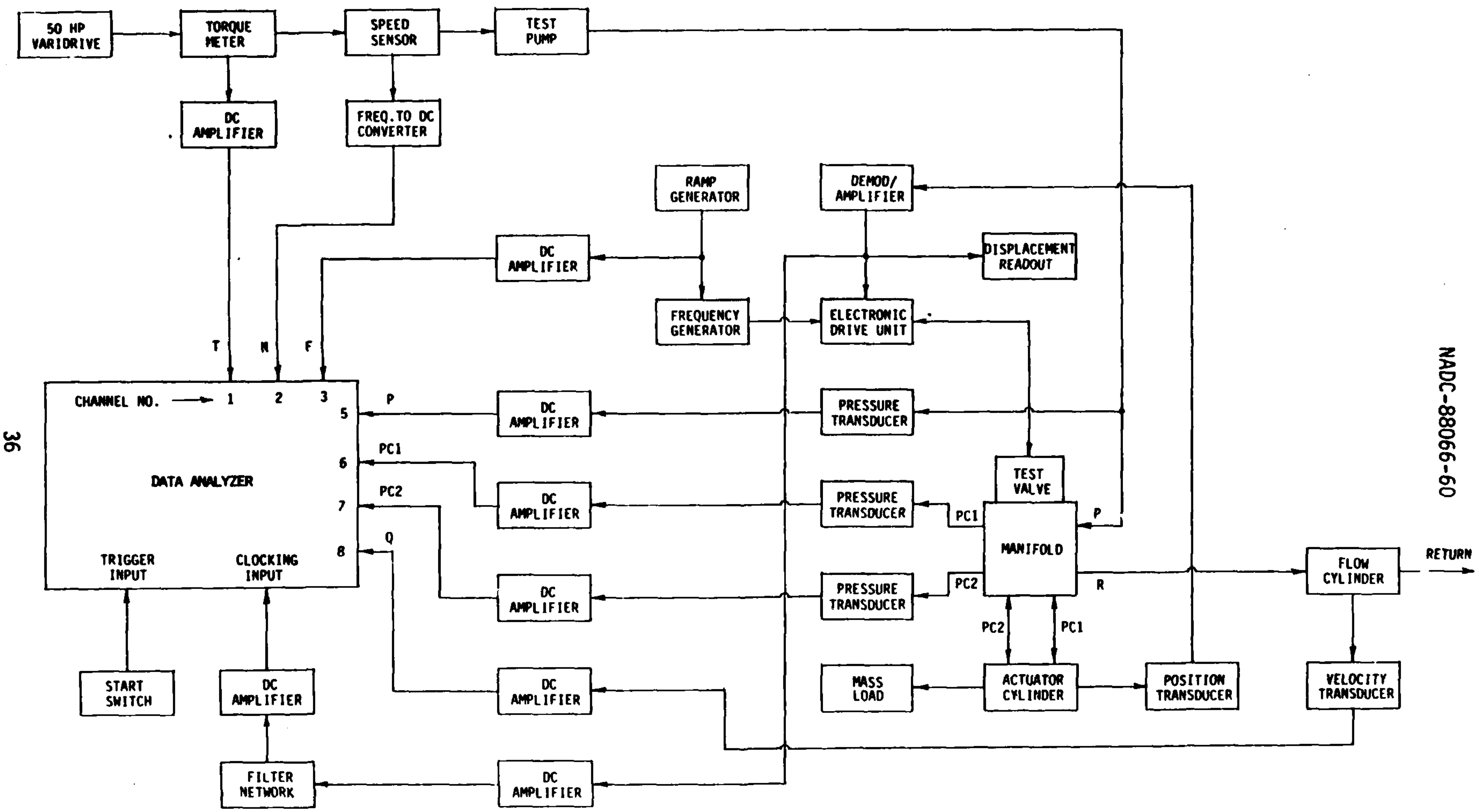


Figure 21. Schematic diagram of energy consumption test instrumentation

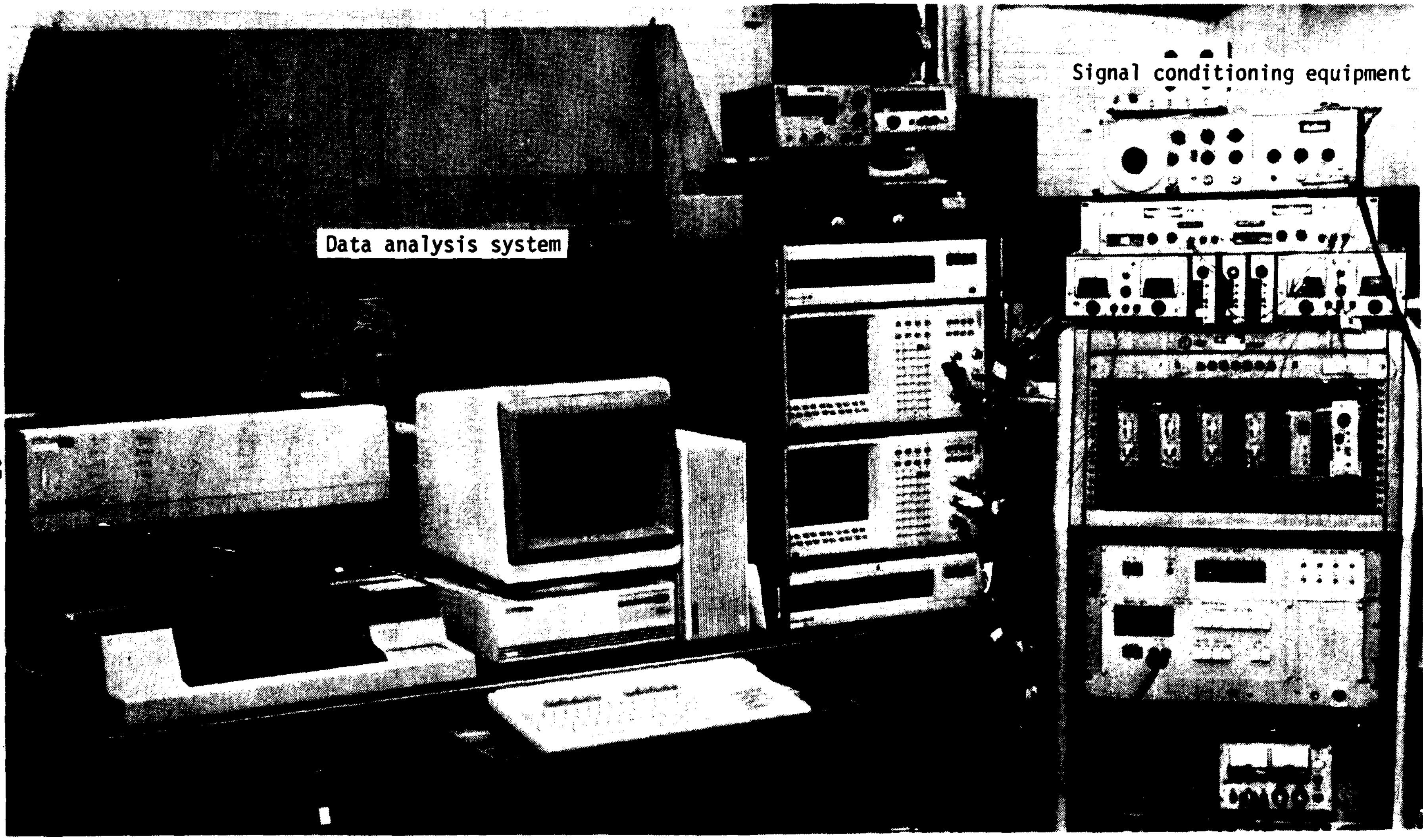


Figure 22. View of energy consumption test instrumentation

TABLE 7. Actuator test parameters and equations

TEST PARAMETERS

CHANNEL NO.	SYMBOL	DESCRIPTION	CALIBRATION FACTOR
1	T	Pump input torque	-400 lb-in/v
2	N	Pump speed	5000 rpm/v
3	F	Actuator frequency	13.93 Hz/v
4	P	Pump discharge pressure	1784 psi/v
5	PC1	Actuator cylinder pressure	1474 psi/v
6	PC2	Actuator cylinder pressure	1456 psi/v
7	Q	Return flow	2.366 gpm/v

TEST EQUATIONS

$$HR_1 = (HP_1 - HP_2) \cdot K_4$$

$$HR_2 = (HP_2 - HP_3) \cdot K_4$$

$$HR_3 = (HP_1 - HP_3) \cdot K_4$$

where,

HR_1 = Pump heat rejection, BTU/min

HR_2 = Actuator heat rejection, BTU/min

HR_3 = Overall heat rejection, BTU/min

HP_1 = Input power to pump, hp

HP_2 = Power delivered by pump, hp

HP_3 = Power delivered by actuator, hp

K_4 = 42.4 BTU/min/hp

now,

$$HP_1 = K_1 \cdot T \cdot N$$

$$HP_2 = K_2 \cdot (P-90) \cdot Q$$

$$HP_3 = K_3 \cdot \left| PC1-PC2 \right| \cdot Q$$

where,

K_1 = 1/63030

K_2 = 0.96/1714

K_3 = 0.98/1714

1.2.2.2 Results

Dynamic Response Tests. Performance plots made during the step input and frequency response tests are presented in Appendix C. Examples of the data are shown on Figures 23 and 24. A summary of this data is given in Table 8. Actuator performance was degraded by valve overlap. Using zero lap data as a baseline, 0.002 in. of valve overlap resulted in the following performance decreases:

Actuator Load	Pressure Level, psi	Step Response	Frequency Response (at -3 db point)
none	4000	-4%	-24%
none	8000	-34%	-21%
mass	4000	-12%	-26%
mass	8000	-22%	-42%

A valve failure occurred during the operational checkout tests. The Bendix T-slot valve was selected for testing first and was being operated with 4000 psi applied. A preliminary frequency sweep from 0.4 Hz was in progress when the motor end cover came off as the 15 Hz level was approached. The end cover is exposed to return pressure and at the time of failure was being subjected to a fluctuating pressure near 400 psi. Examination disclosed no failed parts. The motor housing has a groove in which a circular cross-section ring (2-3/8 in. dia.) is used to retain the motor end cover. This groove was observed to have a sloped outer edge. The retainer ring apparently slid up this slope due to approximately 1000 pounds of force imposed on the end cover by the return pressure. The supplier was notified of the failure. The Bendix valves and electronics were then returned for investigation of the problem. Because of program scheduling constraints, a decision was made to proceed with the planned tests using the E-Systems valves.

STEP RESPONSE

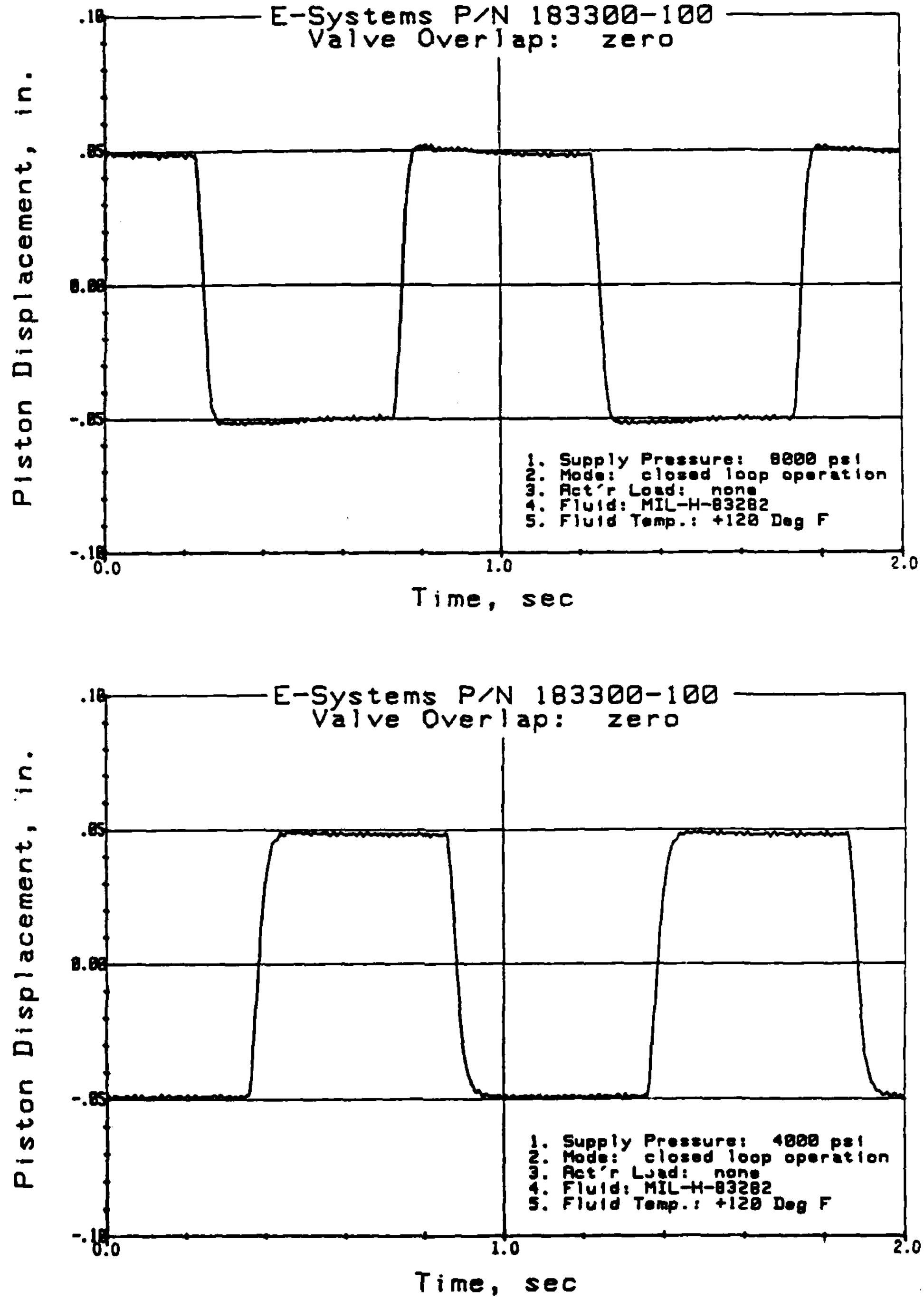
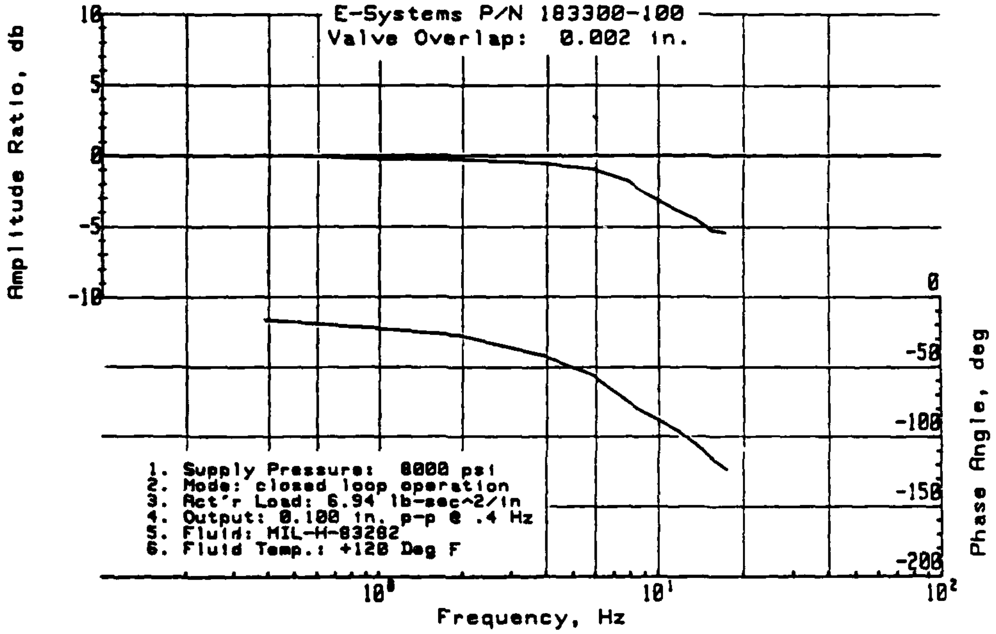


Figure 23. Actuator step response data, E-Systems zero lap valve

FREQUENCY RESPONSE



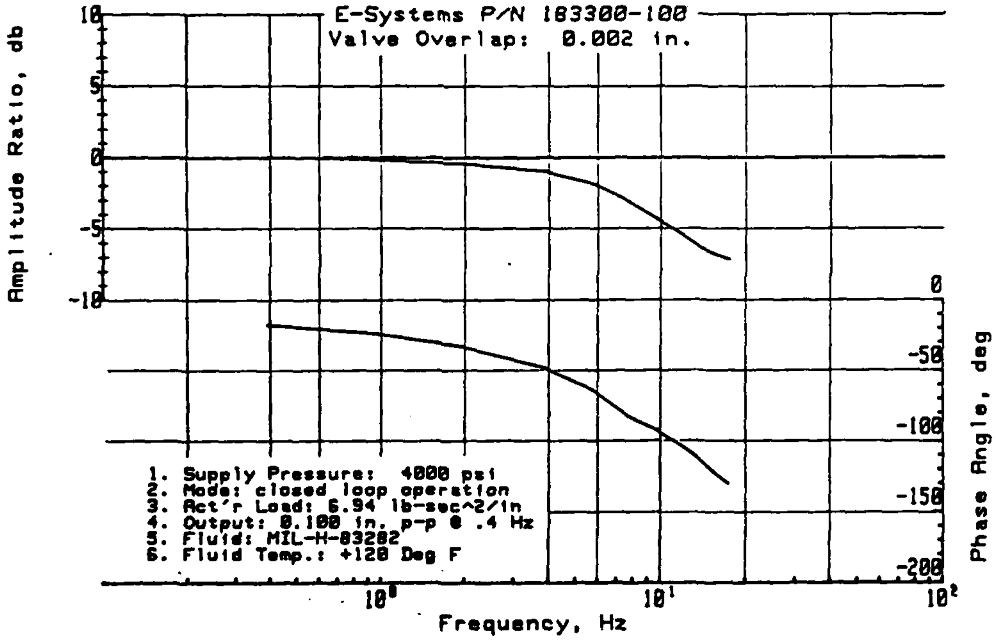


Figure 24. Actuator frequency response data, E-Systems overlapped valve

TABLE 8. Servo actuator dynamic performance, mass load

				STEP RESPONSE	FREQUENCY RESPONSE	
					-3 db POINT	
VALVE	OVERLAP	ACT'R LOAD	PRESSURE	TRANSIT TIME, SEC	FREQUENCY, Hz	PHASE ANGLE, DEG
E-Systems	Zero	None	4000 8000	.086 .062	9.0 11.3	-80 -93
		Mass	4000 8000	.076 .059	10.4 17.0	-86 -112
	.002	None	4000 8000	.089 .083	6.8 8.9	-75 -82
		Mass	4000 8000	.085 .072	7.7 9.8	-82 -88

Pressure Level Switching Tests. Plots of the pressure level switching tests are presented in Appendix C; a plot example is given on Figure 25 A listing of the data is given in Table 9. The results are summarized below:

	Pressure Level Switch	
	4000 to 8000 psi	8000 to 4000 psi
Valve operating time, sec (average)	0.092 (on to off)	0.052 (off to on)
Pressure level switching time, sec (average)		
3400 rpm pump speed	0.139	0.205
5900 rpm pump speed	0.070	0.148
Pressure over/under shoot, psi (average)		
3400 rpm pump speed	410 (over)	290 (under)
5900 rpm pump speed	680 (over)	280 (under)

Significant findings were:

- o Actuator position disturbance was not detectable when the piston was moving and was 0.001 to 0.002 in. when the piston was stationary.

- o Pressure transients that occurred during pressure level switching were well under the acceptable limit of 1600 psi (maximum).

- o The time required for pressure level switchover depended upon pump speed. The average total times were: (valve + pump operating times)

Pump Speed, rpm	4000 to 8000 psi time, sec	8000 to 4000 psi time, sec
3400	0.232	0.259
5900	0.160	0.199

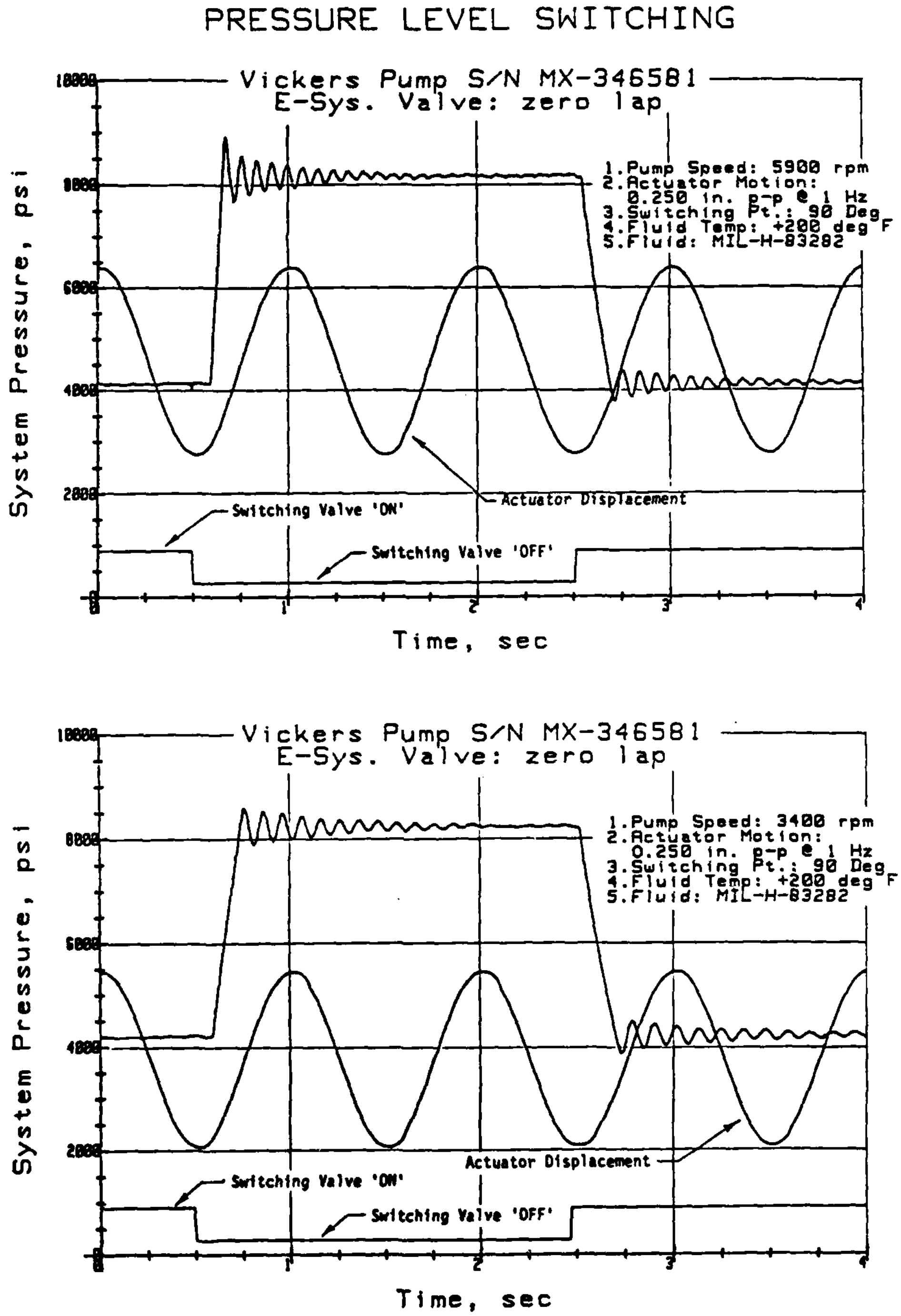


Figure 25. Actuator pressure level switching data, E-Systems zero lap valve

TABLE 9. Pressure level switching data, mass load

E-Systems Valve	Actuator Motion	Pump Speed, rpm	Switch-Over Pt.	Pressure Level Switching: 4000 psi to 8000 psi: Switching Time, sec: Valve	4000 psi to 8000 psi: Switching Time, sec: P. Level	4000 psi to 8000 psi: Pressure Overshoot, psi	8000 psi to 4000 psi: Switching Time, sec: Valve	8000 psi to 4000 psi: Switching Time, sec: P. Level	8000 psi to 4000 psi: Pressure Undershoot, psi
Zero lap	none	3400	-	.103	.130	440	.046	.230	270
		5900	-	.093	.068	730	.048	.155	280
Zero lap	sinusoidal	3400	0°	.096	.129	460	.057	.217	300
			90°	.089	.154	350	.061	.177	340
		5900	0°	.084	.068	740	.051	.148	290
			90°	.095	.069	720	.052	.141	310
Overlap	sinusoidal	3400	0°	.086	.136	450	.054	.212	250
			90°	.093	.146	350	.054	.189	290
		5900	0°	.096	.068	600	.050	.152	240
			90°	.060	.075	620	.033	.144	300

- o Valve lap was not a significant factor in pressure level switchover time.

- o The zero lap valve produced an excellent sine wave motion of the actuator piston; the overlapped valve produced a slightly distorted wave form.

Energy Consumption Tests. Heat rejection plots of the pump, actuator, and system are presented in Appendix C. An actuator plot is shown on Figure 26. A summary of this data is given in Table 10. Significant findings were:

- o Actuator load had a negligible effect on system (total) heat rejection.

- o Energy losses at 4000 psi were approximately 53% of the losses that occurred at 8000 psi.

- o Pump heat rejection accounted for 85 to 90% of the total losses.

- o Valve overlap reduced actuator losses from 15% (zero overlap) of total system losses to 10% (.002 in. overlap).

- o System energy losses were approximately constant over the actuator frequency range of 1 to 3 Hz. Energy consumption began to increase above 3 Hz.

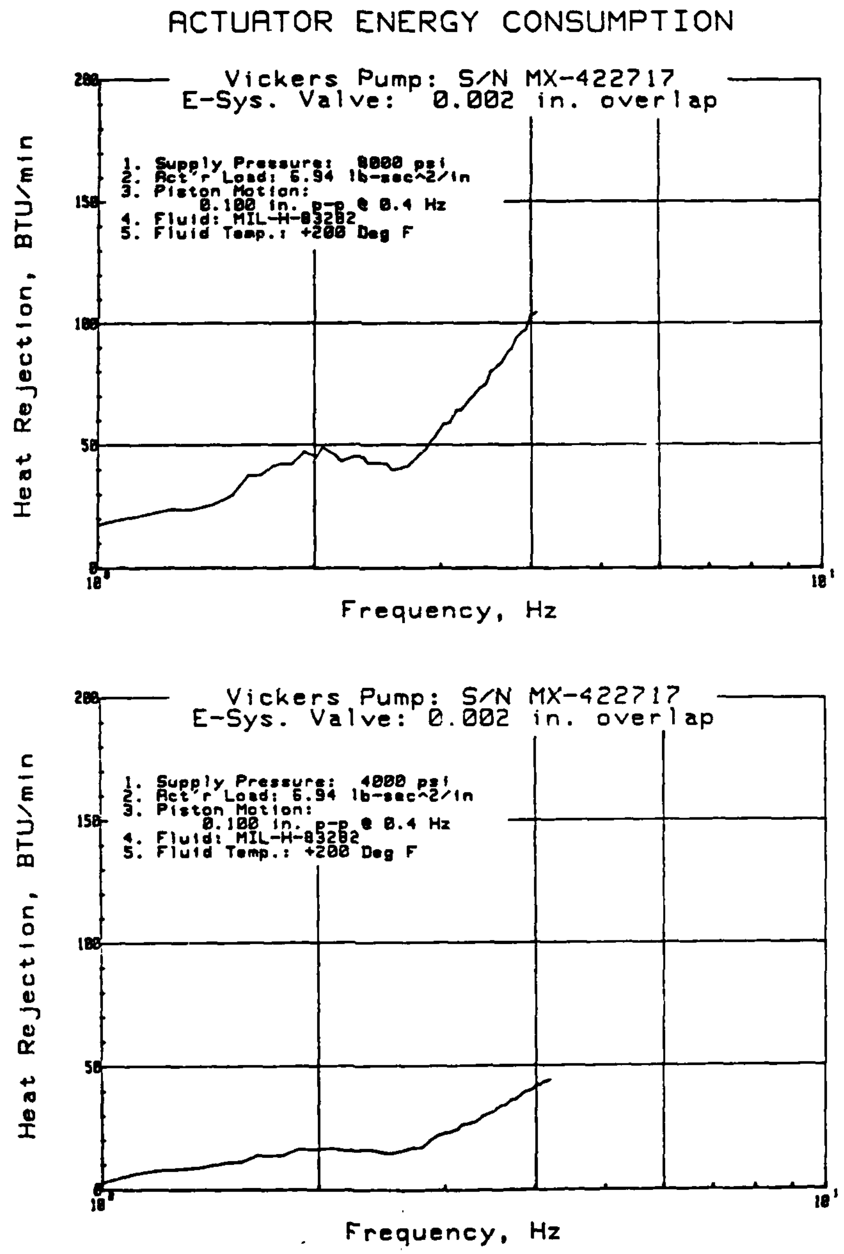


Figure 26. Actuator energy consumption data, E-Systems overlapped valve

TABLE 10. Energy consumption data, mass load

E-SYSTEMS VALVE	ACTUATOR LOAD	PRESSURE LEVEL,psi	*HEAT REJECTION, BTU/min		
			PUMP	ACTUATOR	SYSTEM
Zero lap	none	4000	160	22	185
		8000	298	50	350
Zero lap	mass	4000	162	22	185
		8000	287	56	350
Overlapped	none	4000	162	14	175
		8000	286	32	325
Overlapped	mass	4000	158	16	170
		8000	280	38	317

*Heat rejection data are average values over the frequency range of 1 to 3 Hz. See Appendix C.

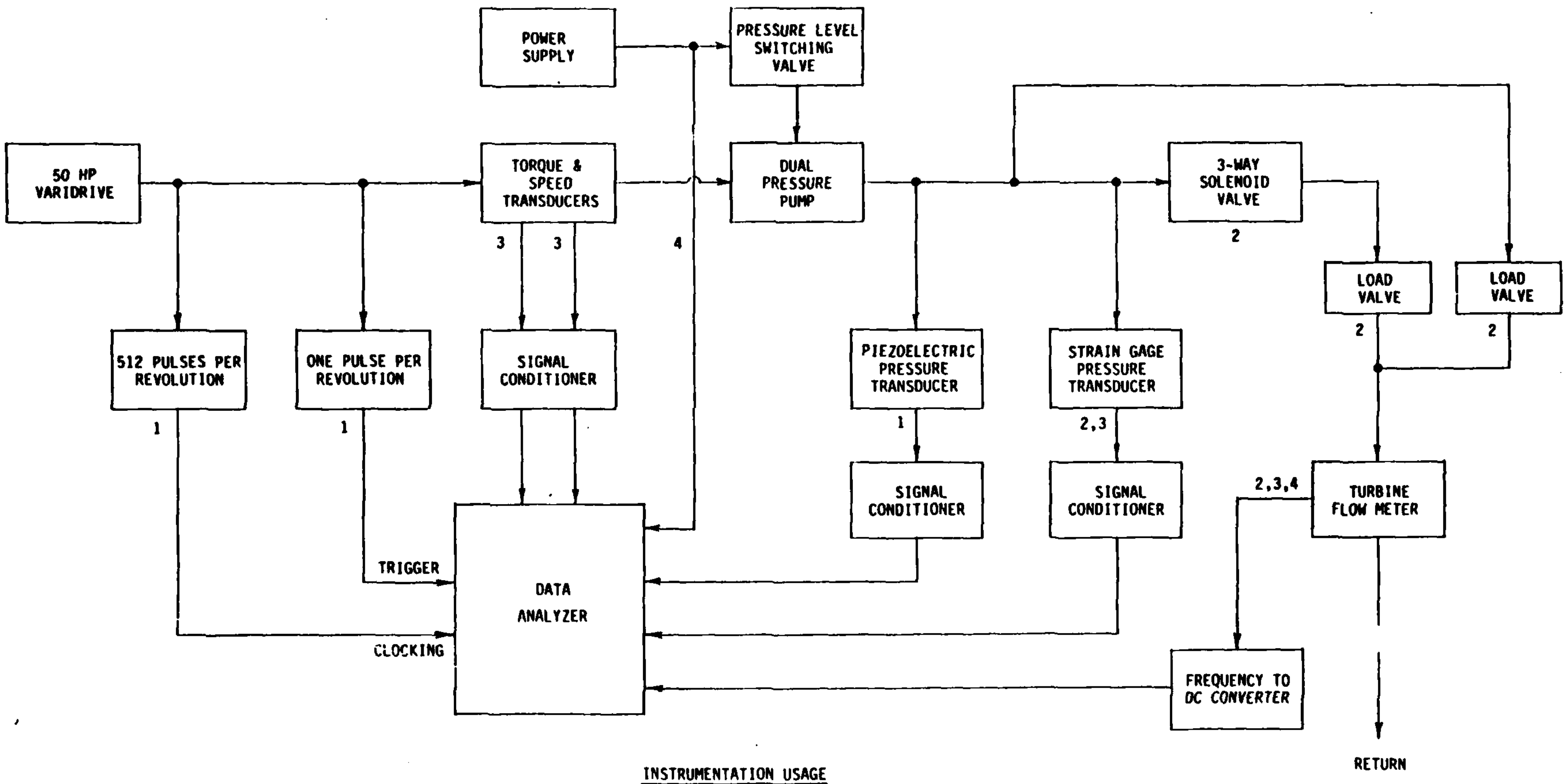
1.2.3 Dual Pressure Pump

1.2.3.1 Procedure. Four different types of tests were conducted: pressure ripple, transient response, pump performance, and pressure level switching. A block diagram of the pump test instrumentation is shown on Figure 27. The pump was mounted on a torque meter attached to a 50 hp varidrive. Controls and monitors for pump speed, discharge pressure, return flow, and input torque were located on a console.

Pressure ripple was measured using a piezoelectric transducer teed into the pressure line at the pump discharge port. The clocking and trigger inputs of the data analyzer were employed to enable measuring pressure fluctuations that occurred during one revolution of the input shaft. The pulsations could thus be correlated to shaft position and individual pump pistons.

Transient response was determined using a strain gage transducer teed into the -8 size discharge line 10 feet from the pump and immediately upstream of a 3-way solenoid valve. System fluid volume at high pressure was approximately 125 cubic inches. Discharge flow was controlled by two pre-set needle valves; one valve was set for 0.5 gpm with the solenoid valve "off", the other was set for 9 gpm with the solenoid valve "on". These flow values were selected based on the requirement to switch from 5% to 90% to 5% of rated flow, reference 3.

Pump operating characteristics in the flow cut-off range of 7700 to 8200 psi were determined in the pump performance tests. Overall efficiency, heat rejection, and discharge pressure were plotted versus discharge flow using the capabilities of the data analyzer. The equations employed are given in Table 11. Data were acquired and results calculated based on pre-selected discharge pressures inserted into the computational program.



INSTRUMENTATION USAGE

Key	Test
1	Pump pressure ripple
2	Pump transient response
3	Pump performance
4	Pressure level switching

Figure 27. Schematic diagram of pump test instrumentation

TABLE 11. Pump test parameters and equations

TEST PARAMETERS

CHANNEL NO.	SYMBOL	DESCRIPTION	CALIBRATION FACTOR
1	P	Pump discharge pressure	1784 psi/v
2	Q	Return flow	4.368 gpm/v
3	T	Pump input torque	-400 lb-in/v
4	N	Pump speed	5000 rpm/v

TEST EQUATIONS

$$EFF = \frac{HP_2}{HP_1} \times 100$$

$$HR = (HP_1 - HP_2) \cdot K_3$$

where,
EFF = Pump overall efficiency, %
HR = Pump heat rejection, BTU/min
HP_1 = Input power to pump, hp
HP_2 = Power delivered by pump, hp
K_3 = 42.4 BTU/min/hp

now,
$$HP_1 = K_1 \cdot T \cdot N$$
$$HP_2 = K_2 \cdot (P - 90) \cdot Q$$

where,
$$K_1 = 1/63030$$
$$K_2 = 0.96/1714$$

The dual pressure level pumps were designed to operate with 90 psig suction pressure. The minimum operating pressure is 45 psig at the pump suction port. Reservoir-to-pump suction line losses and the suction line quick disconnect loss must be factored in; the reservoir should therefore be pressurized to at least 60 psig. The LHS simulator employs bootstrap type reservoirs which provide 90 psig suction pressure when system pressure is 8000 psi. The reservoir provides 45 psig suction pressure when 4000 psi is applied to the bootstrap port. Since this is less than the minimum 60 psig pressure, pump performance at 4000 psi was conducted using a small auxiliary hydraulic power unit to supply 8000 psi to the reservoir bootstrap port.

The effects of switching operating pressure level were determined at pump idle and rated speeds with 0.5 and 5.0 gpm discharge flows. Items of interest were the duration of the switch and pressure transients that occurred. Two parameters were plotted versus time: discharge presssure and voltage applied to the pressure level switching valve. Instrumentation was similar to that used in the actuator pressure level switching tests, Figure 20.

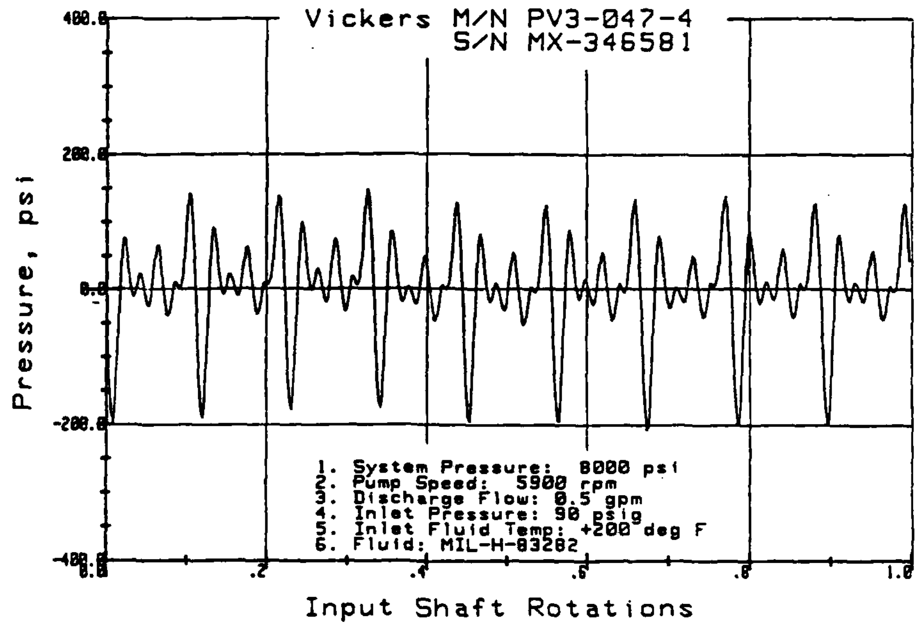
1.2.3.2 Results.

Pump Pressure Ripple. Plots of pump pressure ripple are shown on Figure 28. A summary of results is given below:

Pressure Level, psi	Pressure Ripple, psi p-p
4000	140
8000	320

The ripple occurred at a frequency of 885 Hz and the pumping action of each of the nine pistons was clearly discernable. Ripple magnitude was less than the design allowables, reference Appendix A.

PUMP PRESSURE RIPPLE



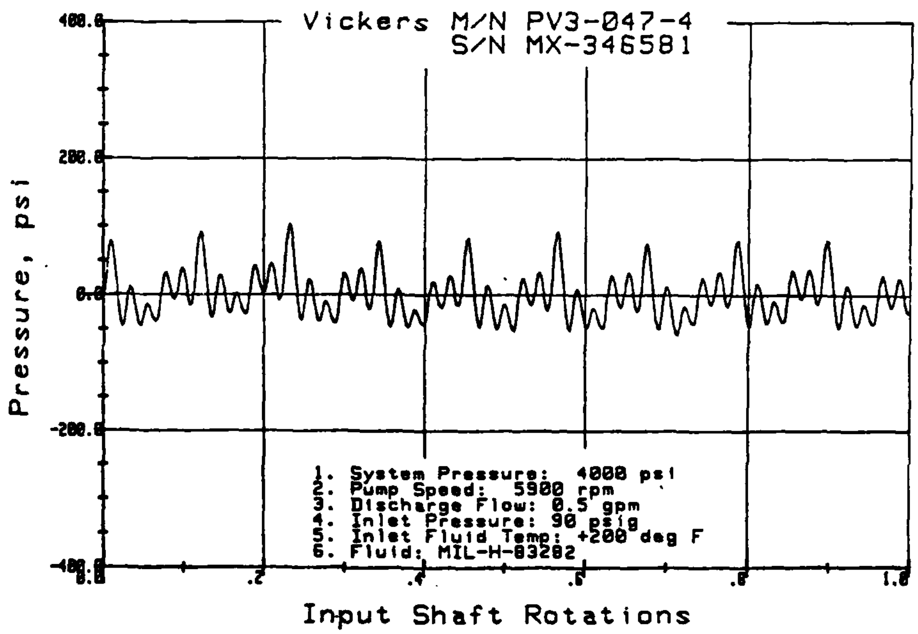


Figure 28. Pump pressure ripple data

Pump Transient Response. Plots of pump transient response are shown on Figure 29. A summary of results is given below:

	Operating Pressure Level	
	4000 psi	8000 psi
5% flow pressure, psi	4125	8130
90% flow pressure, psi	4040	8040
Pressure overshoot at 5% flow, psi	520	510
Pressure undershoot at 90% flow, psi	520	970
Transient response time at 5% flow, sec	.023	.017

The pressure compensator settings for 4000 psi and 8000 psi were slightly out-of-tolerance; this was considered to be a minor discrepancy. Pressure over/under shoot and transient response times were all well within design requirements, reference Appendix A.

Pump Performance. Plots of pump performance are shown on Figure 30. A summary of results is given below:

	Operating Pressure Level	
	4000 psi	8000 psi
Overall efficiency at rated flow, %	83	87
Heat rejection at full cut-off, BTU/min	187	385
Pressure droop, psi/gpm	2	5

Pump performance was considered to be satisfactory except for heat rejection at 8000 psi which was higher than the design goal of 330 BTU/min.

Pressure Level Switching. Plots of pressure level switching are presented in Appendix D. A summary of results is given in Table 12. Pressure level switching time was dependent upon pump speed and flow demand. The design goal of 0.100 sec maximum was met at rated speed for the switch from 4000 to 8000 psi which is the most important operating situation. Pressure transients were well within design allowables. It should be noted that the observed transients do not reflect the effect of change in pump suction pressure which would occur if a non-isolated bootstrap reservoir were used (see Section 1.2.3.1).

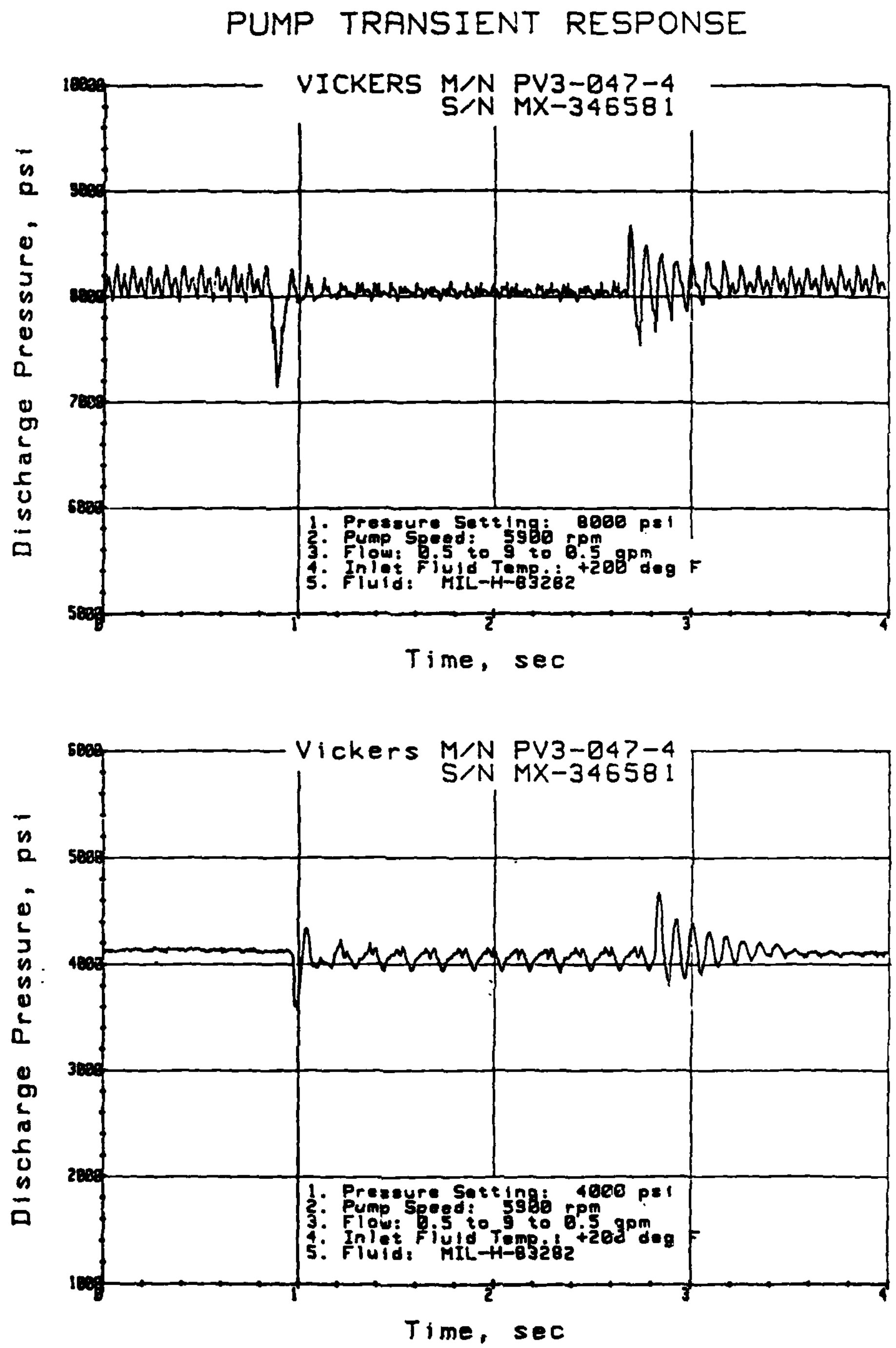
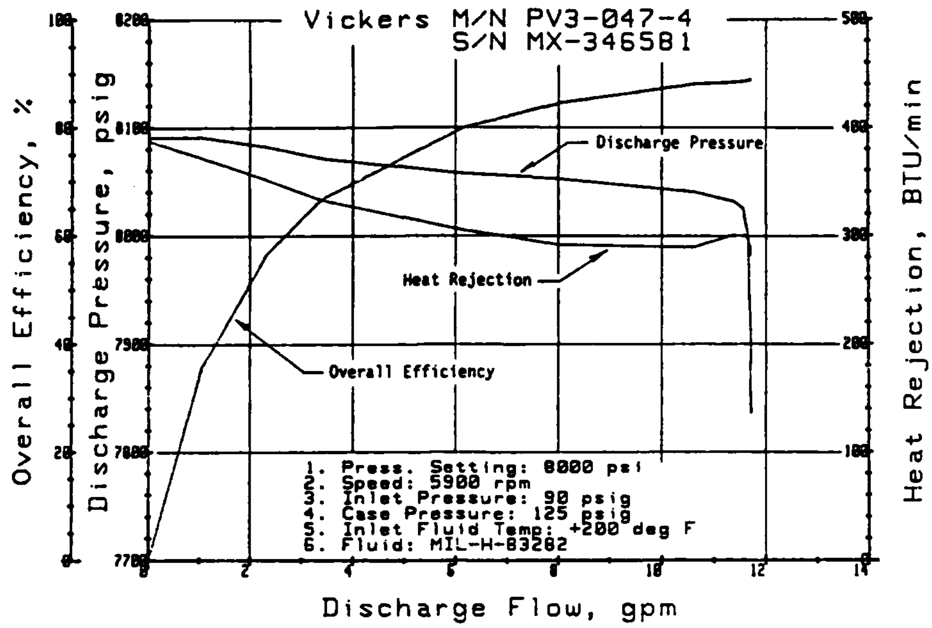


Figure 29. Pump transient response data

PUMP PERFORMANCE



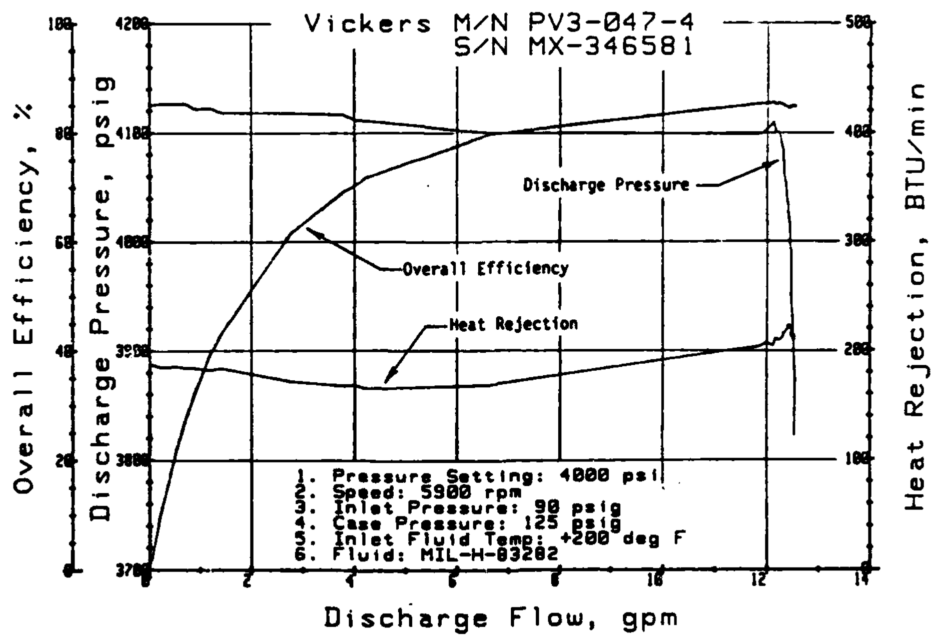


Figure 30. Pump performance data

TABLE 12. Pressure level switching data (pump test)

*Discharge Flow, gpm	Pump Speed, rpm	Pressure Level Switching					
		4000 psi to 8000 psi			8000 psi to 4000 psi		
		Switching Time, Sec		Pressure Overshoot, psi	Switching Time, sec		Pressure Undershoot, psi
		Valve	P. Level		Valve	P. Level	
0.5	3400	.094	.121	450	.056	.179	300
0.5	5900	.075	.069	710	.048	.130	310
5.0	3400	.094	.281	100	.047	.097	530
5.0	5900	.086	.081	310	.038	.083	460

* Flows are at 8000 psi

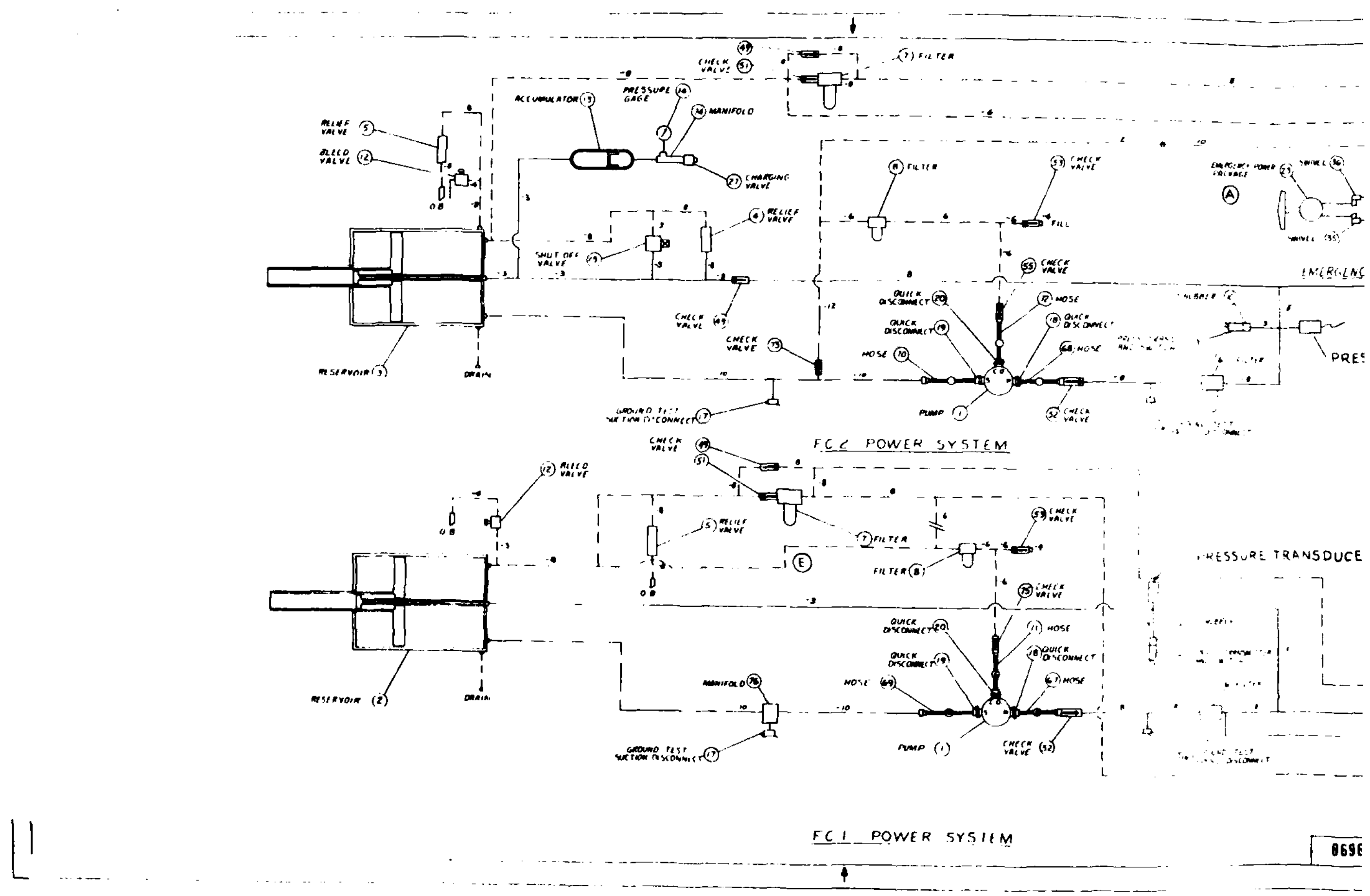
1.2.4 LHS Simulator

1.2.4.1 Procedure. Five different types of tests were conducted: dynamic response, pressure level switching (actuator tests and system tests), pressure ripple, spectrum analysis, and energy consumption. Each test employed different instrumentation and methods of acquiring the data. A schematic diagram of the LHS simulator hydraulic systems is presented as Figure 31.

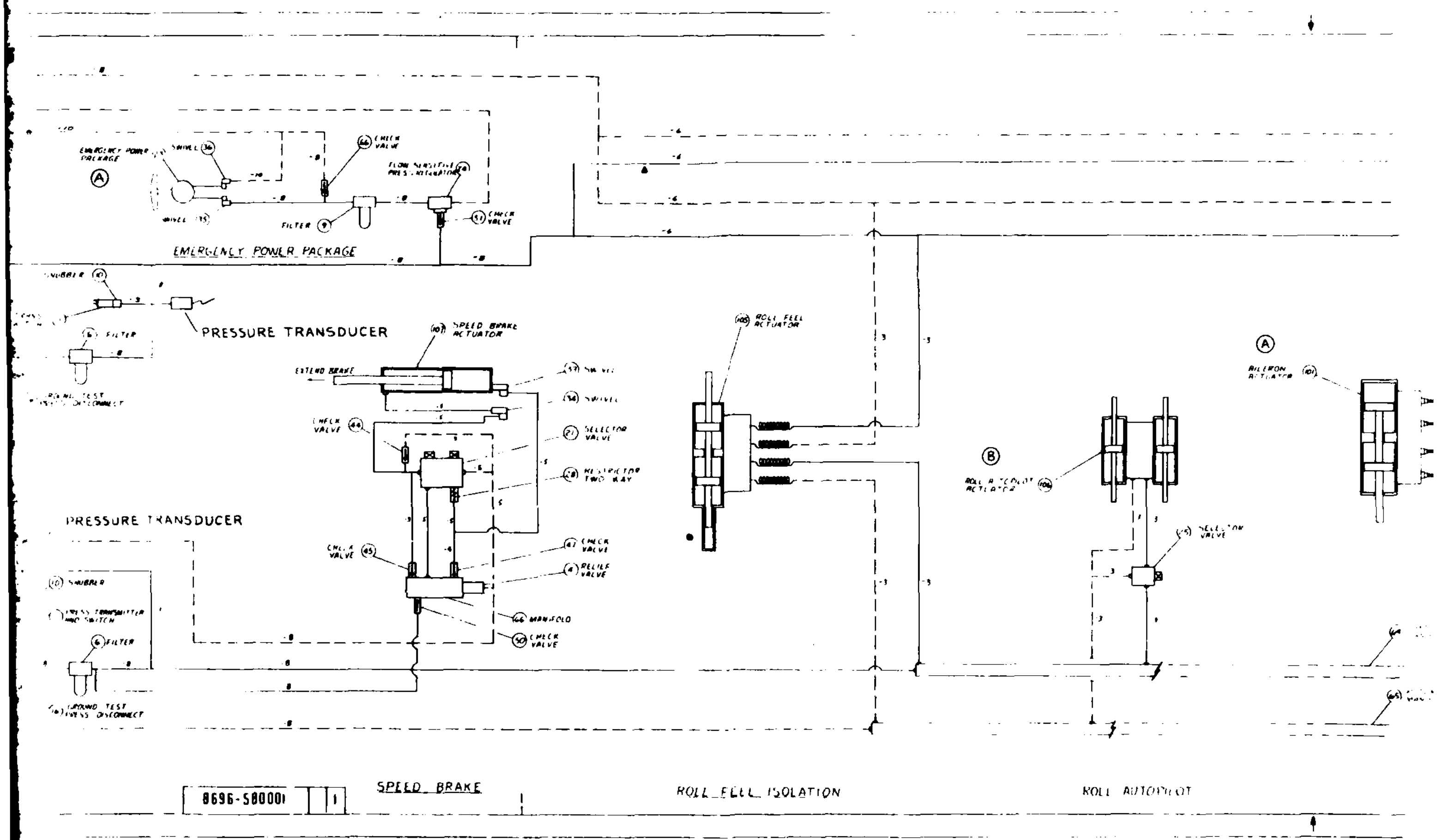
The test servo actuator was mounted in a fixture designed to simulate the kinematics and load of the unit horizontal tail (UHT) installation in the A-7E aircraft, Figure 32 (see reference 1). The UHT actuator load/stroke curve is shown on Figure 33. The actuator was plumbed into FC-1 hydraulic system. FC-1 system was used for all tests; FC-2 system was operated only during the pressure level switching, pressure ripple, and spectrum analysis tests. The dual pressure pump installation is shown on Figure 34.

Dynamic Response Tests. The instrumentation and procedure were similar to those used for the tests conducted on the mass load fixture and described in Section 1.2.2.1. Loop gains were also the same as those used previously and are shown on Figure 19. The step input tests were conducted with a 5000 and 10,000 lb tension load on the servo actuator for 4000 and 8000 psi operation, respectively. The frequency response tests were performed with a 5000 lb tension load on the actuator for both 4000 and 8000 psi operation.

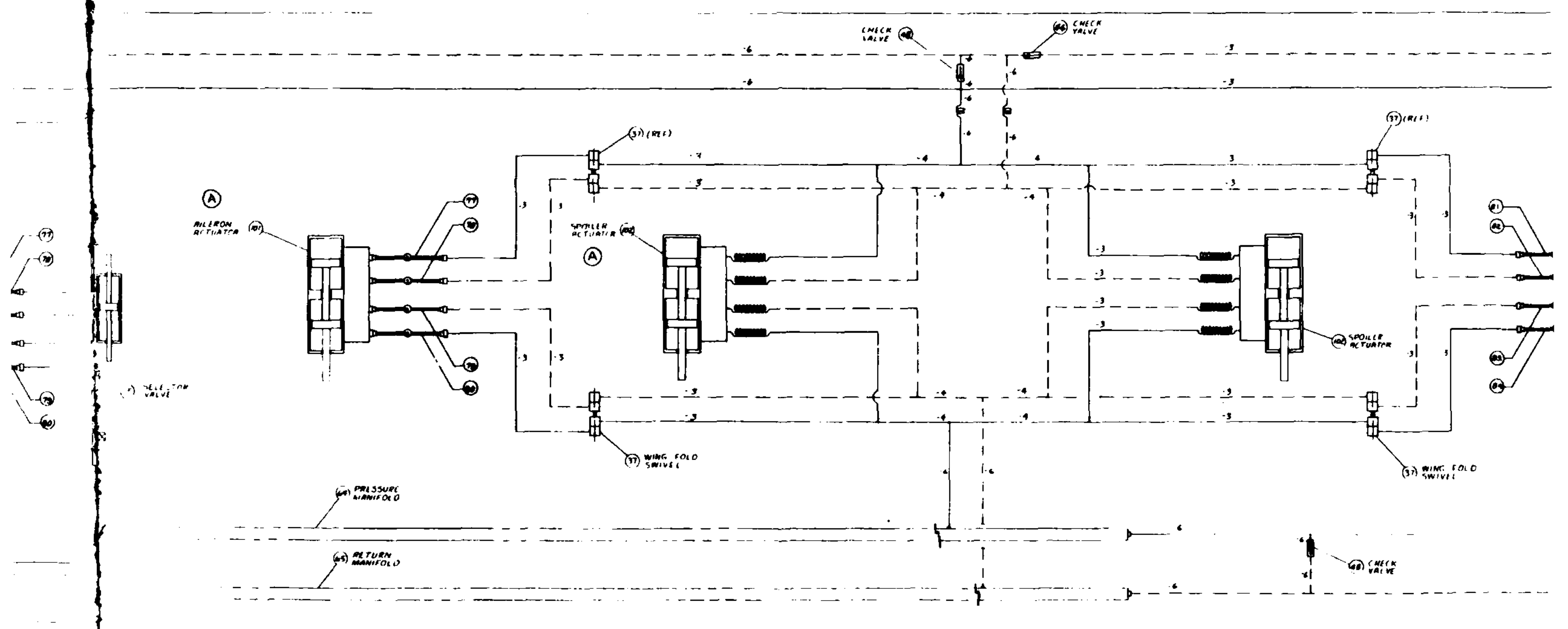
Pressure Level Switching (Actuator Tests). The instrumentation and procedure were similar to those used for the tests conducted on the mass load fixture and described in Section 1.2.2.1. A 5000 lb tension load was imposed on the actuator during the pressure level switching. This load is approximately 20% of the maximum actuator output (at 8000 psi) and occurs when the piston is retracted 0.62 in. from neutral, see Figure 33. Data were collected with the actuator motionless and with the piston moving sinusoidally. Pressure level switching was performed at two locations on the output sine wave -- 0^{o} and 90^{o}.



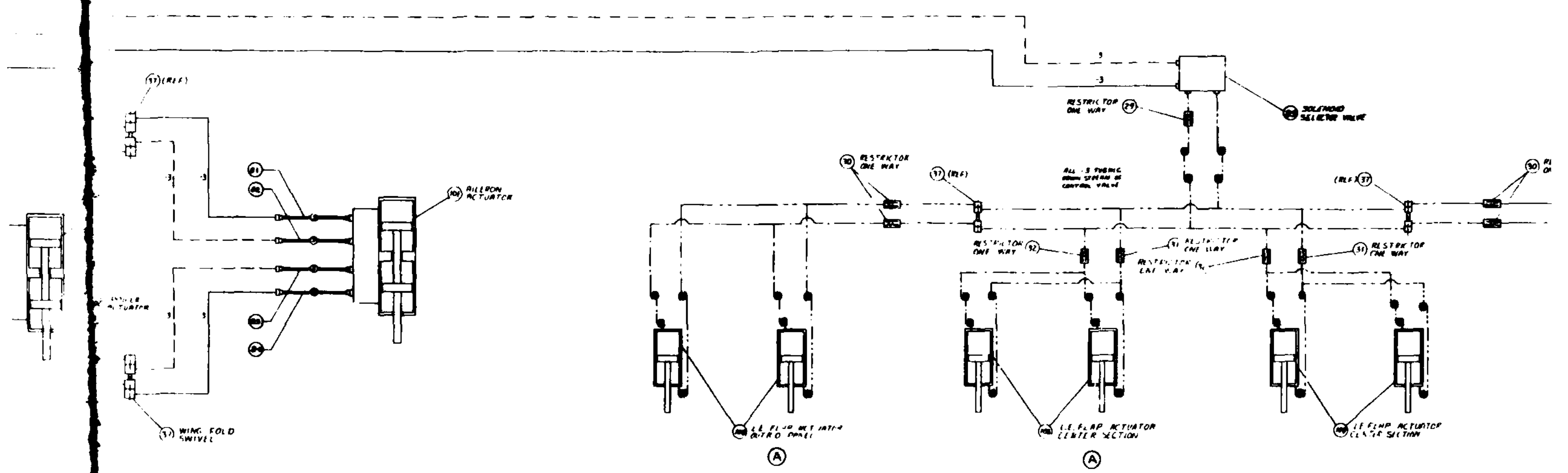
FC 2 POWER SYSTEM
FC 1 POWER SYSTEM
ACCUMULATOR (13)
PRESSURE GAGE (16)
MANIFOLD (14)
CHARGING VALVE (27)
RELIEF VALVE (5)
BLEED VALVE (12)
RESERVOIR (3)
RESERVOIR (2)
DRAIN
SHUT OFF VALVE
RELIEF VALVE (4)
CHECK VALVE (49)
CHECK VALVE (75)
FILTER (7)
FILTER (8)
CHECK VALVE (53)
FILL
CHECK VALVE (55)
QUICK DISCONNECT (20)
QUICK DISCONNECT (19)
QUICK DISCONNECT (18)
HOSE (12)
HOSE (10)
HOSE (68)
PUMP (1)
CHECK VALVE (52)
GROUND TEST SUCTION DISCONNECT (17)
MANIFOLD (76)
HOSE (69)
HOSE (67)
HOSE (11)
EMERGENCY POWER PACKAGE
SWIVEL
EMERGENCY
PRESSURE TRANSDUCE



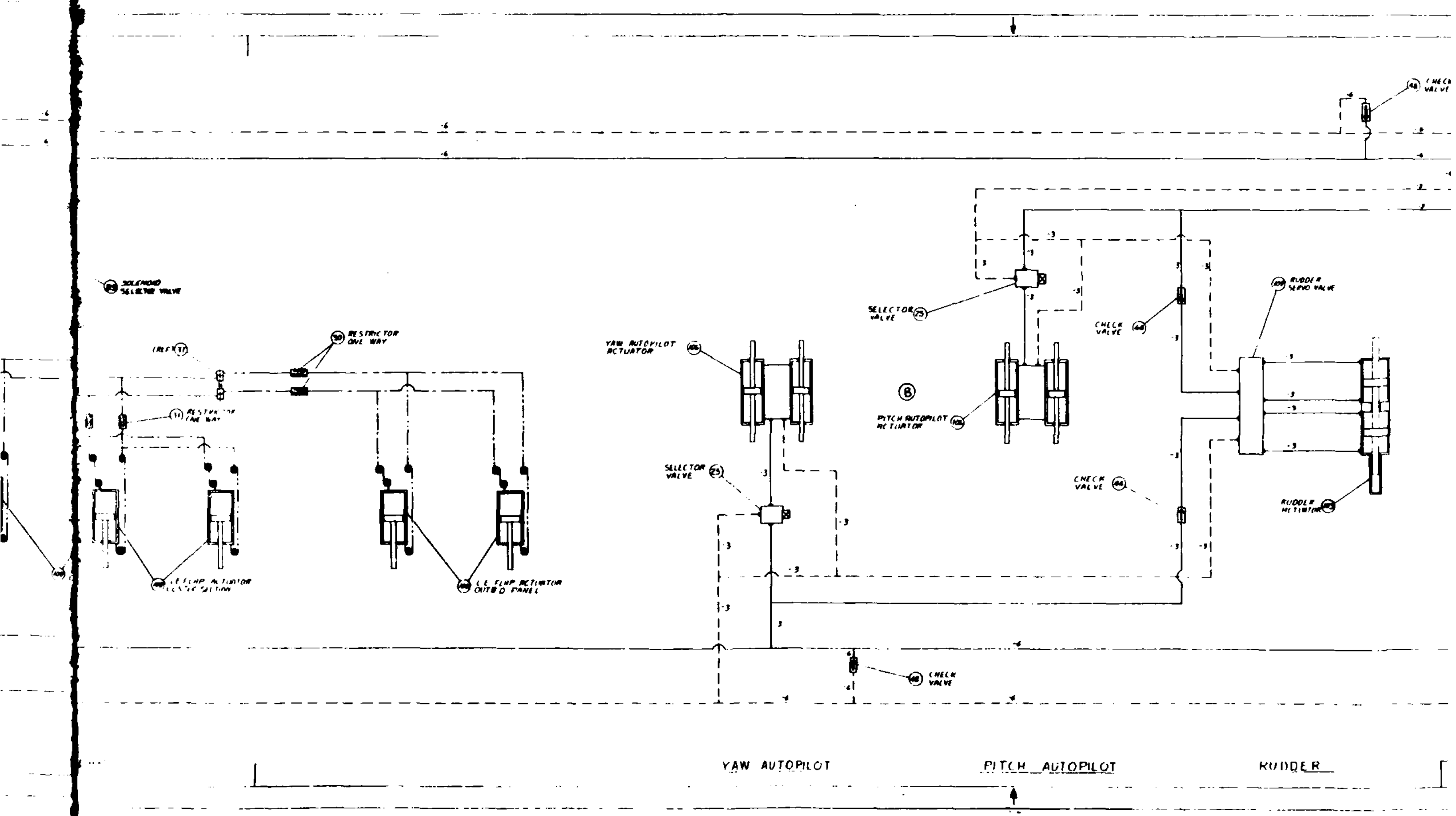
EMERGENCY POWER PACKAGE
PRESSURE TRANSDUCER
PRESSURE TRANSDUCER
SPEED BRAKE ACTUATOR
EXTEND BRAKE
ROLL FEEL ACTUATOR
AILERON ACTUATOR
SELECTOR VALVE
CHECK VALVE
RELIEF VALVE
MANIFOLD
FILTER
8696-580001
1
SPEED BRAKE
ROLL FEEL ISOLATION
ROLL AUTOPILOT



8696-580001 | 1 | AILERON AND SPOILER



8696-580001 | 1 | LEADING EDGE FLAP



CHECK VALVE
SOLENOID SELECTOR VALVE
RESTRICTOR ONE WAY
SELECTOR VALVE
YAW AUTOPILOT ACTUATOR
CHECK VALVE
RUDDER SERVO VALVE
PITCH AUTOPILOT ACTUATOR
CHECK VALVE
RUDDER ACTUATOR
L.E. FLAP ACTUATOR OUTB'D PANEL
CHECK VALVE
YAW AUTOPILOT
PITCH AUTOPILOT
RUDDER

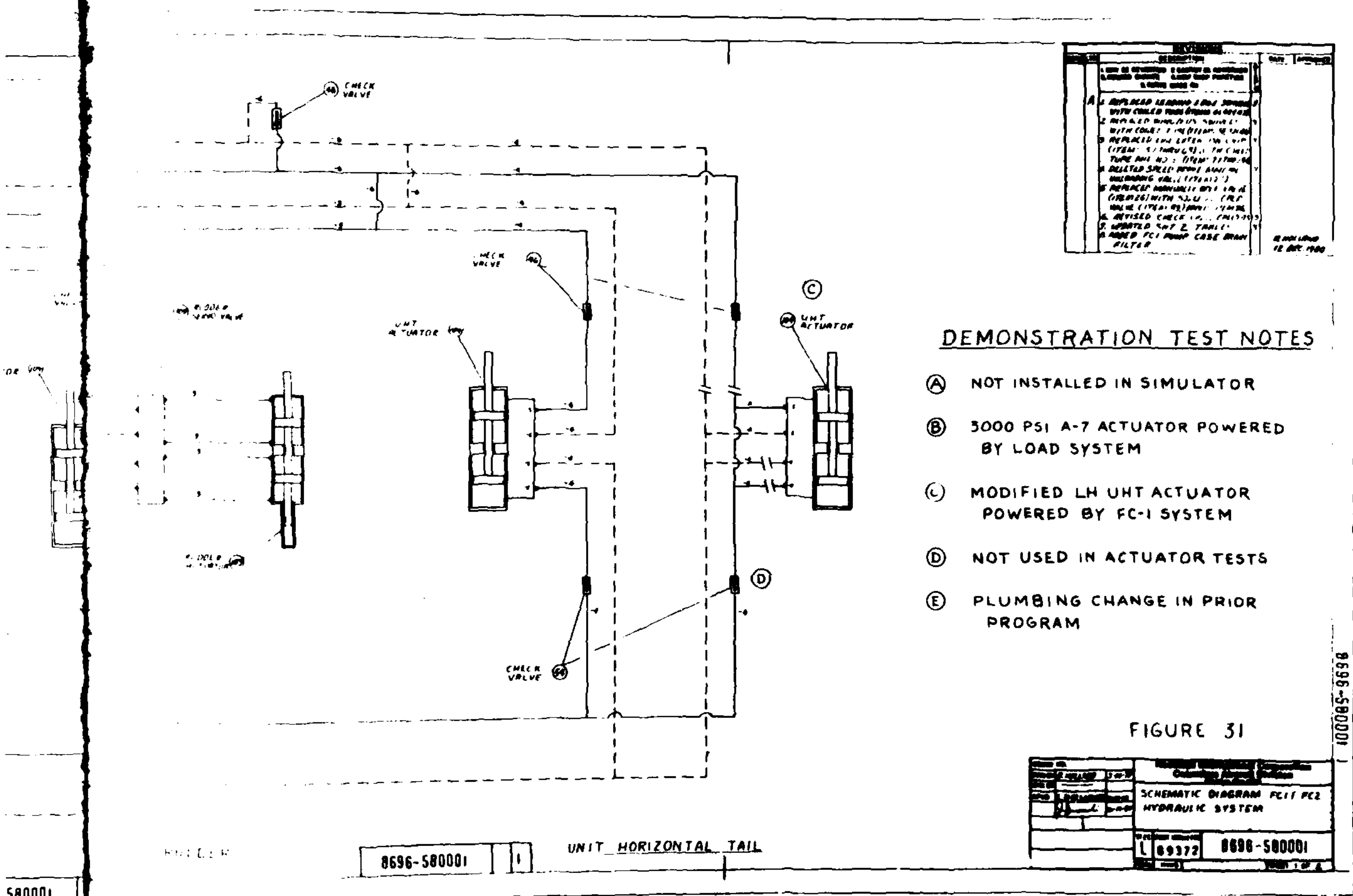

CHECK VALVE
CHECK VALVE
UHT ACTUATOR
UHT ACTUATOR
CHECK VALVE
DEMONSTRATION TEST NOTES
Ⓐ NOT INSTALLED IN SIMULATOR
Ⓑ 3000 PSI A-7 ACTUATOR POWERED BY LOAD SYSTEM
Ⓒ MODIFIED LH UHT ACTUATOR POWERED BY FC-1 SYSTEM
Ⓓ NOT USED IN ACTUATOR TESTS
Ⓔ PLUMBING CHANGE IN PRIOR PROGRAM
SCHEMATIC DIAGRAM FC1 & FC2 HYDRAULIC SYSTEM
69372
8696-580001
8696-580001
UNIT HORIZONTAL TAIL

FIGURE 31

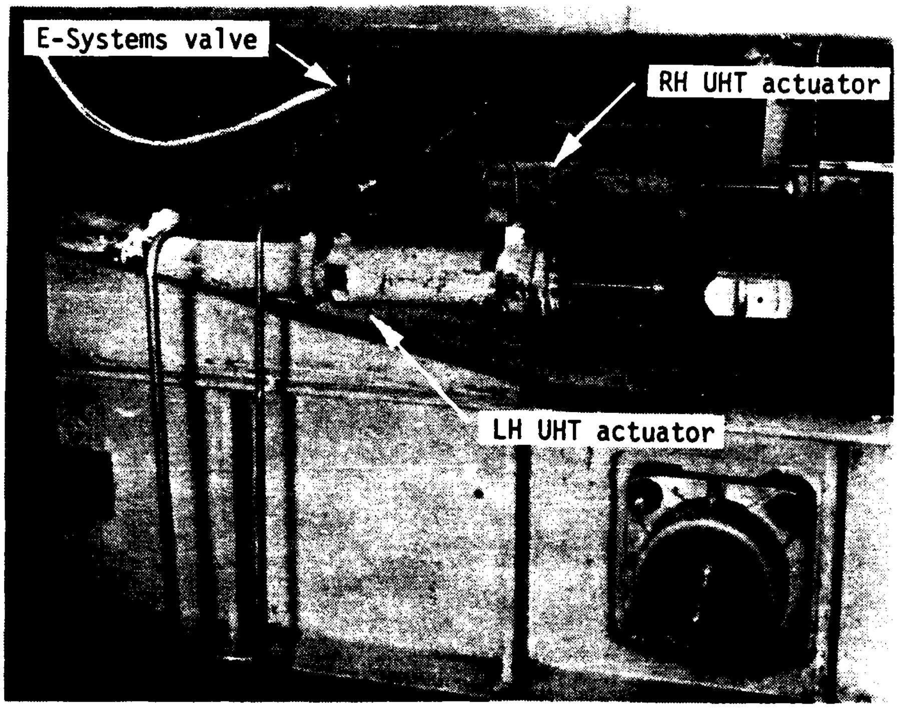


Figure 32. Test actuator in LH UHT load module

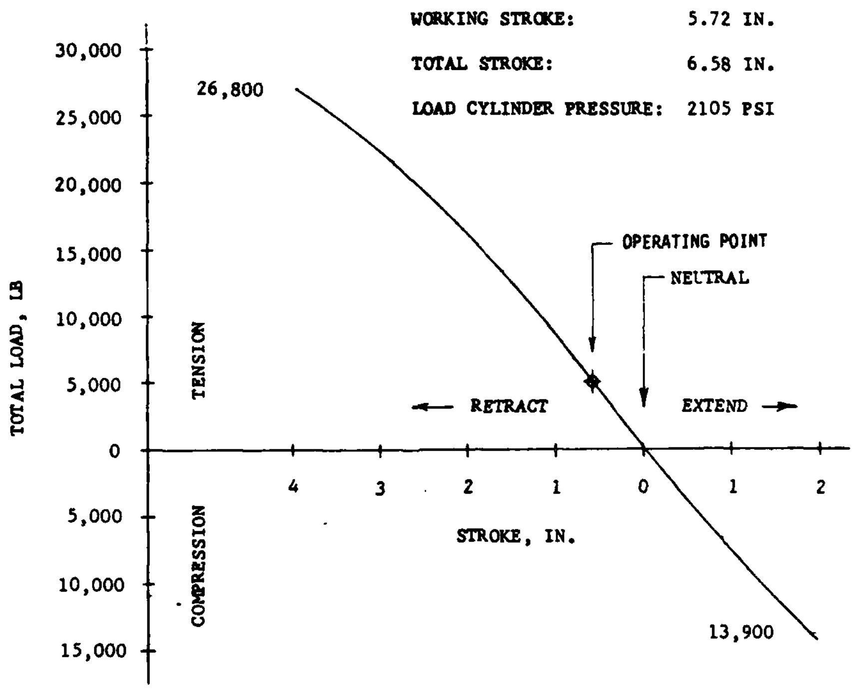


Figure 33. UHT actuator load/stroke curve

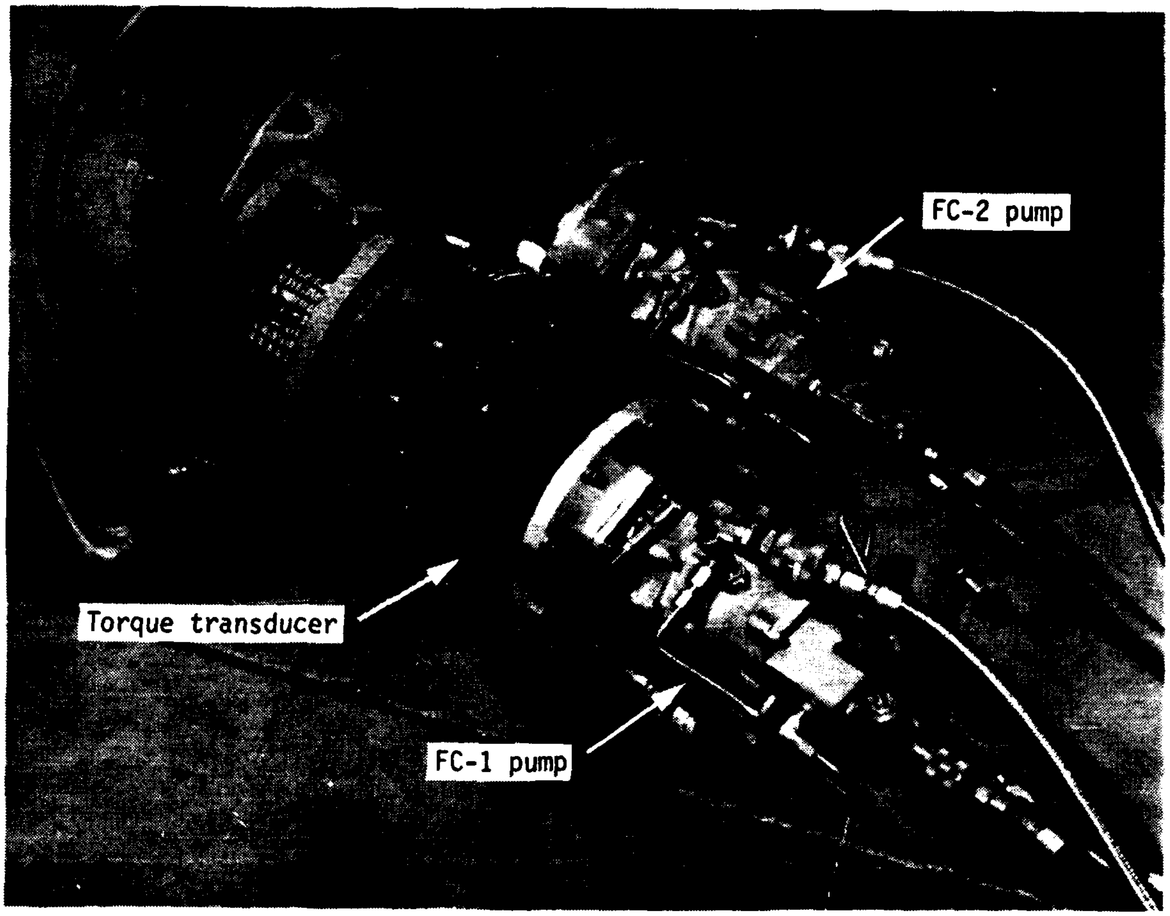


Figure 34. FC-1 and FC-2 system dual pressure pumps

Pressure Level Switching (System Tests). FC-1 and FC-2 hydraulic systems were run concurrently during the pressure level switching tests. The primary flight control actuators on the simulator were operated in a manner similar to that employed during mission/profile endurance cycling (see reference 1). Actuator load/stroke cycling modes used were 2%, 10%, and 50%. Although the same cycling mode was used for the pitch, roll, and yaw axes during a given test, cycling was independent and not in phase. Operation of the LH UHT actuator was different from the other actuators during the 50% mode tests; the unit was intentionally operated near flow saturation. Actuator cycling was sinusoidal in form and at 3, 1, and 0.5 Hz for the 2%, 10%, and 50% operating modes, respectively. The LH UHT actuator (alone) was cycled at 1 Hz during the 50% mode. Pressure level switching was performed at the 90^{0} point on the output sine wave of the LH UHT actuator (actuator load was approximately 5000 lb tension at this point in time).

Four parameters were recorded during the pressure level switching tests:

1. Voltage to the pressure level switching valves mounted on the pumps.
2. LH UHT actuator piston position.

3. & 4. System pressure immediately downstream of the pressure line filters in FC-1 and FC-2 systems (see Figure 31).

Data were collected for the following conditions:

Test	System	Actuator Cycling Mode	Pressure Level Switching
1,2,3	FC-1 & FC-2	2%, 10%, 50%	8000 to 4000 to 8000 psi
4,5,6	FC-1	2%, 10%, 50%	8000 to 4000 to 8000 psi
	FC-2		not switched, 8000 psi maintained

Pump speed was 5900 rpm and pump inlet fluid temperature was $+200^{0}$F.

Pressure Ripple. Pump pressure ripple was measured in FC-1 and FC-2 systems using a piezoelectric transducer teed into the pressure line at the pump discharge port. The clocking and trigger inputs of the data analyzer were employed to enable measuring pressure fluctuations that occurred during one revolution of the pump input shaft (see Figure 27). All flight control actuators in the simulator were motionless and at their null position during the pressure ripple measurements. Pump speed was 5900 rpm and pump inlet fluid temperature was +200°F. Pump suction pressure was maintained at 90 psig for operation at 8000 psi and 4000 psi. A small auxiliary hydraulic power supply was used to apply 8000 psi to the bootstrap port on the FC-1 reservoir when the 4000 psi pressure level tests were conducted. The emergency reservoir pressurization circuit provided in FC-2 system (see Figure 31) maintained 8000 psi on the FC-2 reservoir bootstrap port when system operating pressure was 4000 psi.

Spectrum Analysis. The frequency and amplitude components of the pressure fluctuations in the pump discharge lines were determined using the FFT (Fast Fourier Transform) capability of the data analysis system. A piezoelectric transducer teed into the pressure line 1.25 inches from the pump discharge port was used to provide the pressure signal. This location was selected as likely to provide high amplitude components. Standing waves downstream could possibly produce higher amplitude components, but time did not permit searching for anti-nodes (their location and amplitude vary with pump speed). The data provided by the analyzer are RMS values or 0.707 of the peak value.

The test was conducted by collecting FFT data while performing pump speed scans. Ten scans were made between 3400 and 4400 rpm, and between 4900 and 5900 rpm. This data was presented in map form (3-dimensional cascade display). The scan judged to have the highest amplitude components on the map was then selected for detail analysis which was presented on a 2-dimensional frequency domain plot. All flight control actuators in the simulator were motionless and at their null position during the scans. Data were obtained for FC-1 and FC-2 systems operating at 4000 and 8000 psi.

Energy Consumption Tests. The instrumentation and procedure were similar to those used for the tests conducted with the mass load and described in Section 1.2.2.1. Two flows were planned to be used in the data analysis calculations: LH UHT actuator return flow (measured by a positive displacement flow cylinder) and FC-1 system return flow (measured by a turbine meter). The flow cylinder has a velocity transducer that produces a DC voltage proportional to flow. A frequency-to-DC converter was used with the turbine meter to provide a DC voltage proportional to flow. Unfortunately, the slight phase lag that is inherent in all frequency-to-DC converters was sufficient that the data collected from the turbine flowmeter was not valid. The procedure for acquiring the data and performing the energy consumption calculations was therefore modified.

The energy consumption tests were conducted with the LH UHT actuator operating over a frequency range of 1 to 4 Hz while all other actuators in FC-1 system were stationary at null. The total leakage through the control valves on the simulator actuators plus the combined leakage of other system components such as solenoid valves, check valves, and relief valve were essentially constant during a test run. This tare leakage was measured with system pressures of 4000 psi and 8000 psi and a pump inlet fluid temperature of +200°F. These power losses were then inserted as constants into the energy consumption calculations. Since this total leakage was less than 0.3 gpm at 8000 psi, it had negligible effect on pump heat rejection, no affect on actuator heat rejection, and permitted the test to be conducted without using a system return flow meter. All tests were conducted with a pump speed of 5400 rpm. (This speed was used in the energy consumption tests performed with the mass load, see Section 1.2.2).

1.2.4.2 Results.

Dynamic Response Tests. Performance plots made in the step input and frequency response tests are presented in Appendix E. A summary of this data is given in Table 13. Actuator performance was degraded by valve overlap. Using zero lap data as a baseline, 0.002 in. of valve overlap resulted in the following decreases:

TABLE 13. Servo actuator dynamic performance, force load

VALVE	OVERLAP	ACT'R LOAD, LB	PRESSURE, PSIG	STEP RESPONSE TRANSIT TIME, SEC	FREQUENCY RESPONSE -3 db POINT FREQUENCY, Hz	FREQUENCY RESPONSE -3 db POINT PHASE ANGLE, DEG
E-Systems	Zero	5000	4000	.078	5.3	-68
		10000	8000	.055	7.6	-77
	.002	5000	4000	.079	3.3	-63
		10000	8000	.064	4.4	-68

Actuator Load	Pressure Level, psi	Step Response Time	Frequency Response (at -3 db point)
5000	4000	-1%	-38%
10,000	8000	-16%	-42%

Pressure Level Switching (Actuator Tests). Plots of the pressure level switching tests are presented in Appendix E and sample data is shown on Figure 35. A listing of pertinent data is given in Table 14. The results are summarized below:

	Pressure Level Switch (Actuator piston moving)	
	4000 to 8000 psi	8000 to 4000 psi
Valve operating time, sec (average)	.107	.053
Pressure level switching time, sec. (average)		
3400 rpm pump speed	.118	.464
5900 rpm pump speed	.075	.471
Pressure over/under shoot, psi (average)		
3400 rpm pump speed	440 (over)	102 (under)
5900 rpm pump speed	752 (over)	88 (under)

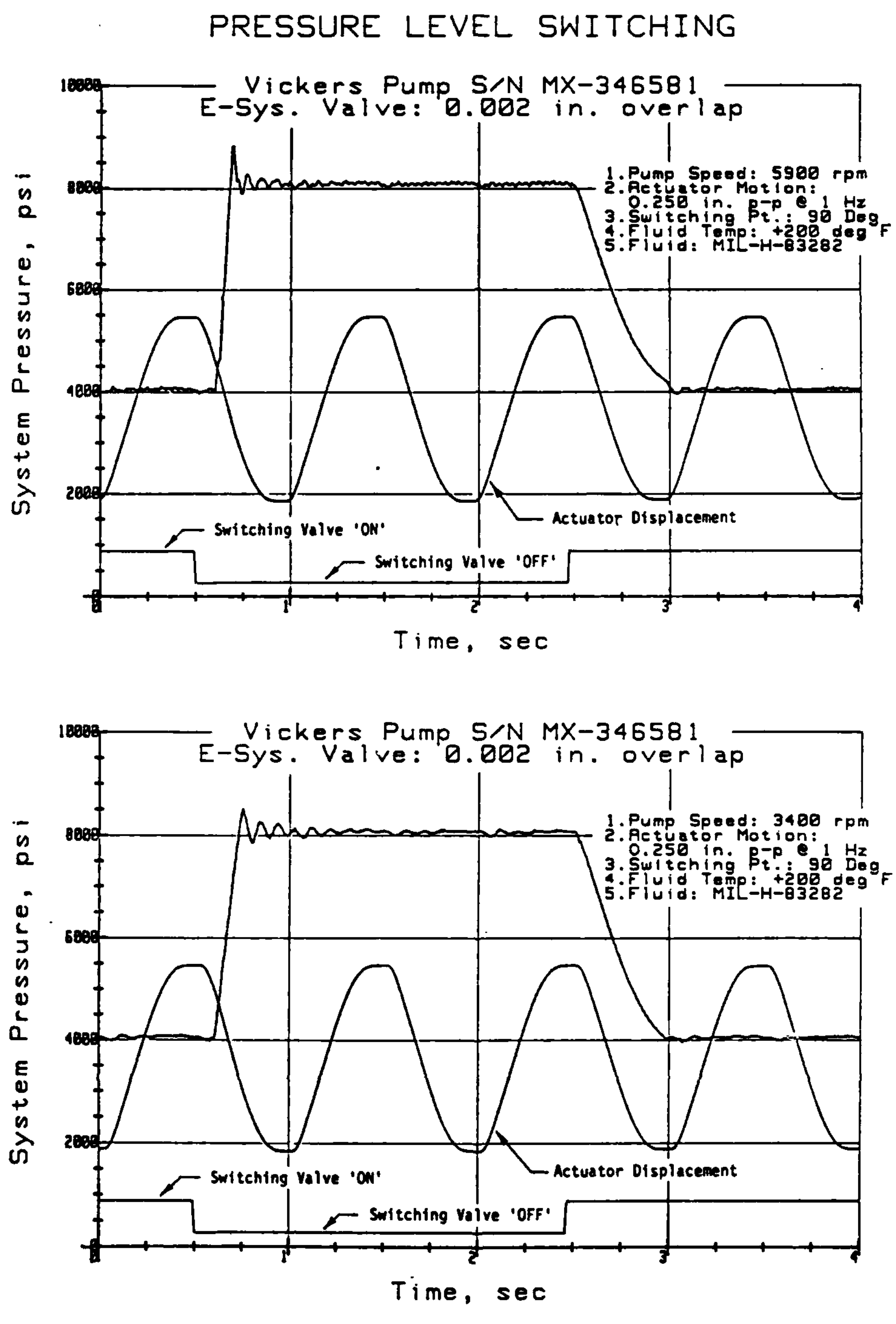


Figure 35. Pressure level switching data (actuator test), E-Systems overlapped valve

TABLE 14. Pressure level switching data (actuator test)

E-Systems Valve	Actuator Motion	Pump Speed, rpm	Switch-Over Pt.	Pressure Level Switching*					
				4000 psi to 8000 psi			8000 psi to 4000 psi		
				Switching Time, sec		Pressure Overshoot, psi	Switching Time, sec		Pressure Undershoot, psi
				Valve	P. Level		Valve	P. Level	
Zero lap	none	3400	-	.099	.121	470	.044	.697	40
		5900	-	.126	.074	770	.059	.677	40
Overlap	none	3400	-	.108	.105	450	.050	1.019	40
		5900	-	.124	.068	840	.055	1.323	0
Zero lap	sinusoidal	3400	0°	.105	.116	500	.046	.467	80
			90°	.104	.120	380	.055	.435	90
		5900	0°	.098	.075	830	.057	.467	100
			90°	.104	.075	660	.059	.423	50
Overlap	sinusoidal	3400	0°	.114	.105	510	.055	.495	150
			90°	.106	.133	370	.054	.460	90
		5900	0°	.116	.075	830	.057	.493	90
			90°	.110	.076	690	.042	.503	110

*Actuator load: 500J lb tension

Significant findings were:

- o With a 5000 lb tension load applied (20% of maximum output at 8000 psi), actuator position disturbance was not detectable when the piston was moving and was less than 0.005 in. when the piston was stationary.
- o Pressure transients that occurred during pressure level switching were well under the acceptable limit of 1600 psi (maximum).
- o The 8000 to 4000 psi switch-over time was significantly higher when: 1) the actuator piston was stationary; and 2) valve overlap was used.
- o The average total times were: (valve + pump operating times, actuator piston moving)

Pump Speed, rpm	4000 to 8000 psi time, sec	8000 to 4000 psi time, sec
3400	0.226	0.517
5900	0.182	0.525

- o The zero lap valve produced an excellent sine wave motion of the actuator piston; the overlapped valve produced a slightly distorted wave form.

Pressure Level Switching (System Tests). Plots of the pressure level switching tests are presented in Appendix E and sample data is shown on Figure 36. A listing of pertinent data is given in Table 15. The total time to switch from 4000 psi to 8000 psi was fairly constant and averaged 0.173 sec.

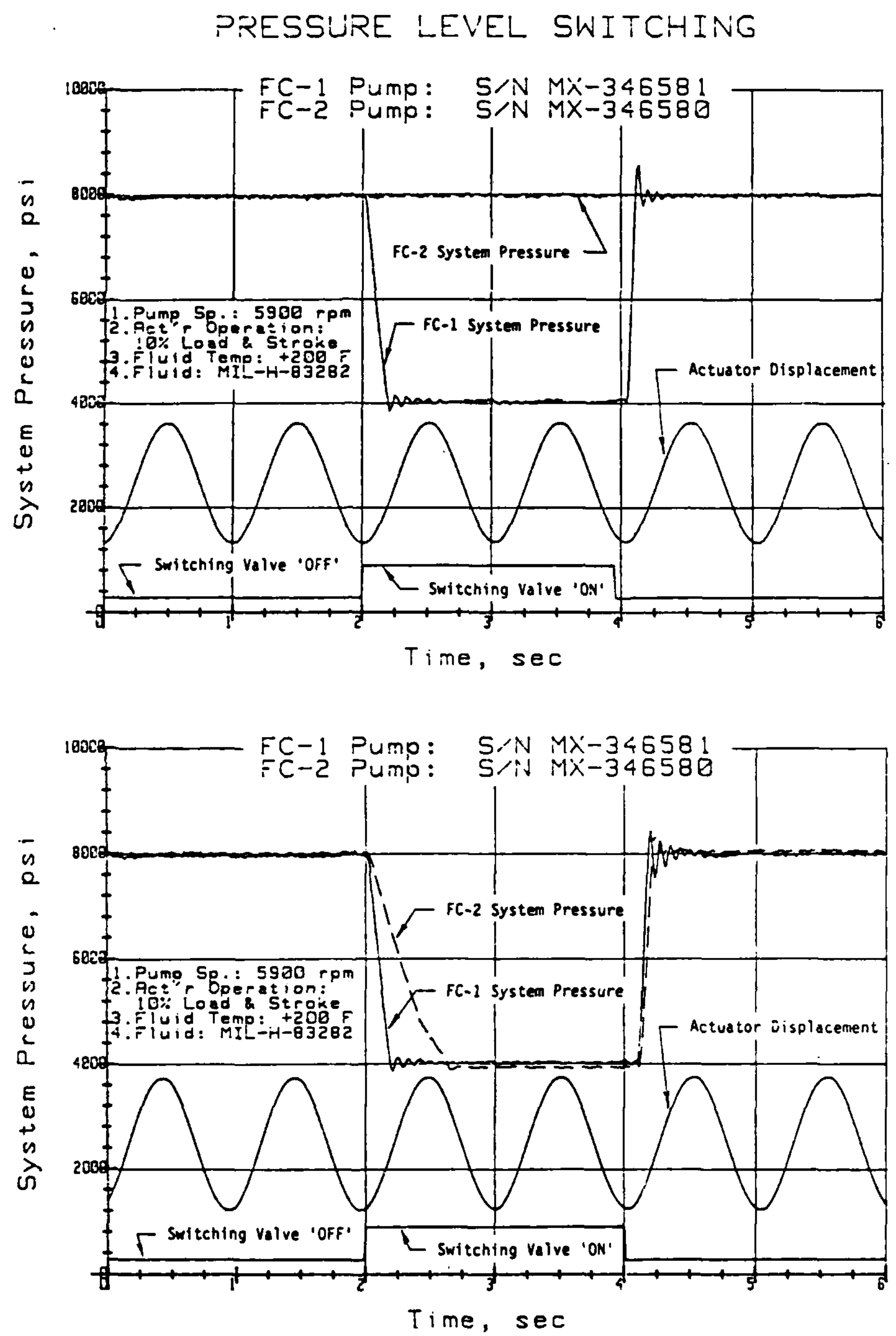


Figure 36. Pressure level switching data (system test), E-Systems zero lap valve

TABLE 15. Pressure level switching data (system test)

System Switched	Operating Mode	Pressure Level Switching							
		8000 psi to 4000 psi				4000 psi to 8000 psi			
		*Total Switching Time, sec		Pressure Undershoot, psi		*Total Switching Time, sec		Pressure Overshoot, psi	
		FC-1	FC-2	FC-1	FC-2	FC-1	FC-2	FC-1	FC-2
FC-1 & FC-2	2%	.326	1.102	65	20	.172	.172	440	370
	10%	.197	.650	170	40	.169	.205	425	240
	50%	.124	.410	255	40	.174	.190	235	150
FC-1	2%	.324	-	113	-	.145	-	550	-
	10%	.202	-	170	-	.162	-	540	-
	50%	.109	-	250	-	.165	-	200	-

*Switching valve operating time plus pressure level switching time

The total time to switch from 8000 psi to 4000 psi depended upon flow demand and system internal leakage; elapsed time ranged from 0.109 sec to 1.102 sec. FC-1 system had higher leakage than FC-2 due to the yaw AFCS actuator in FC-1. A check valve installed in the pressure line at the RH UHT actuator also contributed to the pressure bleed down time because 8000 psi pressure was trapped in the actuator and could drop to 4000 psi only by throttling or leaking through the actuator control valve. (The check valve was removed from the LH UHT actuator pressure line to facilitate conducting the energy consumption tests.)

Pressure transients resulting from switching FC-1 and FC-2 from 8000 psi to 4000 psi to 8000 psi were minor and well within acceptable limits. No problems occurred when FC-1 pressure level was switched to 4000 psi while FC-2 was held at 8000 psi. Overall, the results of the system pressure level switching tests were excellent.

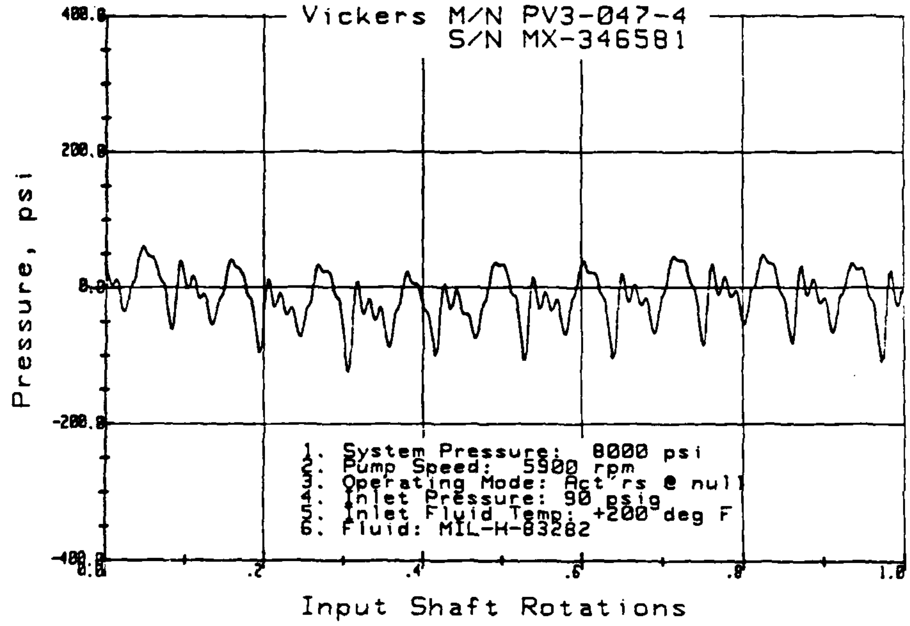
The effects of overloading the E-Systems valve were examined during the pressure level switching tests and can be seen on the plots for the 50% mode presented in Appendix E. Operation at 8000 psi was near flow saturation as evidenced by slight distortion in the actuator piston output wave form. At 4000 psi, the actuator output wave form was severely distorted and decreased in amplitude. This was considered to be typical performance degradation for the given operating conditions.

Pressure Ripple. Time history plots of FC-1 and FC-2 pressure ripple near the pumps are shown in Appendix E and an example is presented on Figure 37. A summary of results is given below:

System	Pressure Level, psi	Pressure Ripple, psi p-p
FC-1	4000	124
	8000	155
FC-2	4000	191
	8000	249

Ripple magnitude was well under the design allowables at both 4000 psi and 8000 psi operating levels.

PUMP PRESSURE RIPPLE, FC-1 SYS.



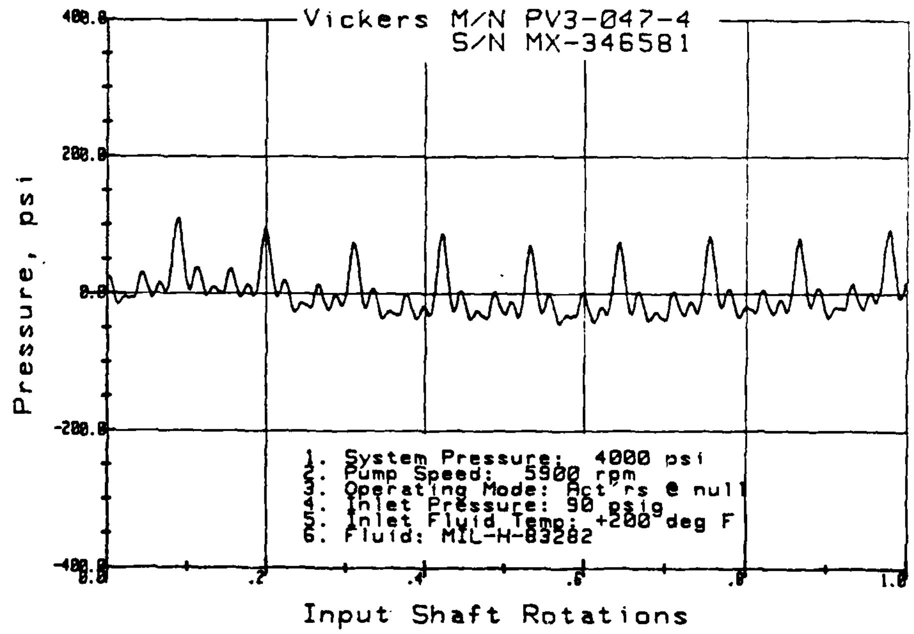


Figure 37. Pump pressure ripple data, FC-1 system

Spectrum Analysis. Two types of spectrum analysis plots are presented in Appendix E: 1) 3-dimensional pump speed scan map; and 2) 2-dimensional detail examination of the map scan containing the highest magnitude components. An example of this data is shown on Figure 38. The test results are summarized in Table 16. The pressure component at the fundamental frequency ($Hz = \frac{rpm \times 9}{60}$) was generally the largest, and all were under 65 psi (RMS peak) or 184 psi p-p. No spurious frequencies were found, and no harmonics occurred over 5000 Hz. The pressure dynamics resulting from pump ripple appeared to be well behaved at both the 4000 psi and 8000 psi operating levels.

Energy Consumption Tests. Heat rejection plots of FC-1 pump, LH UHT actuator, and FC-1 system are presented in Appendix E. Sample data is shown on Figure 39. A summary of this data is given in Table 17. Significant findings were:

- Actuator load had a negligible effect on system (total) heat rejection.
- Energy losses at 4000 psi averaged 47% of the losses that occurred at 8000 psi.
- Pump heat rejection accounted for about 80% of the total losses.
- Valve overlap reduced actuator losses approximately 20%.
- System energy losses gradually increased as frequency increased and reached a maximum between 3 and 4 Hz.

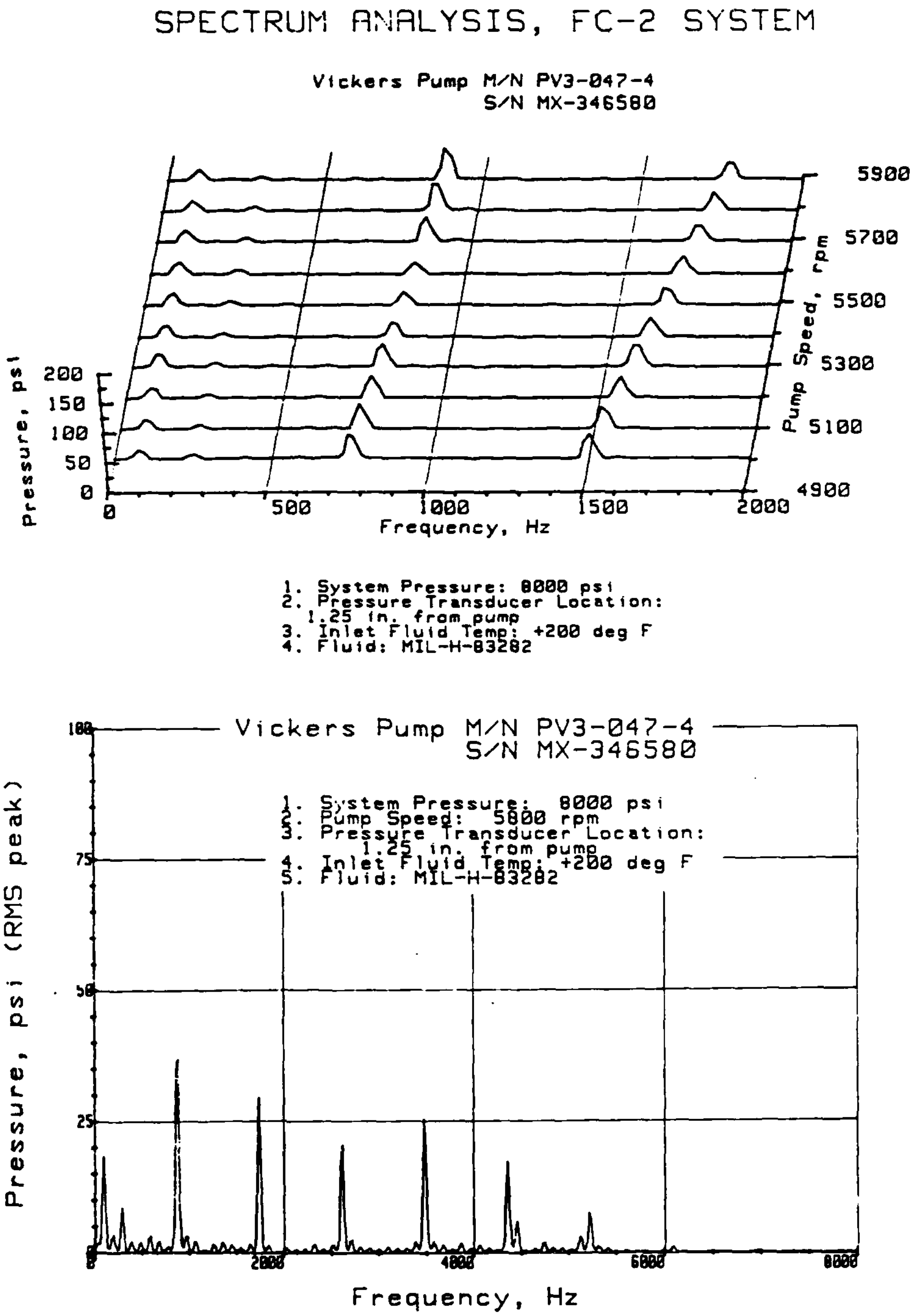


Figure 38. Spectrum analysis data, FC-2 system

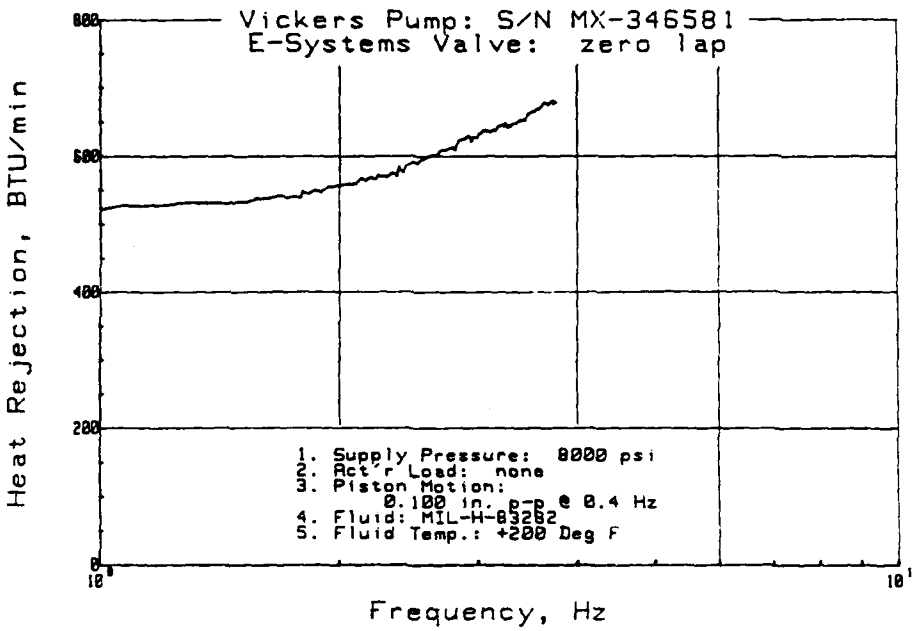
TABLE 16. Spectrum analysis data

System	Pressure Level, psi	*Pump Speed, rpm	Pressure Ripple Components, psi (RMS peak) Fundamental	1st Harmonic	2nd Harmonic
FC-1	4000	3600	14	16	12
		5800	26	22	23
	8000	3900	70	38	20
		5200	55	60	32
FC-2	4000	3900	33	36	14
		5800	38	33	32
	8000	3900	62	31	7
		5800	36	28	20

*Speed selection based on speed scan map data.

Selected speed produced the highest magnitude pressure pulsation components

SYSTEM ENERGY CONSUMPTION



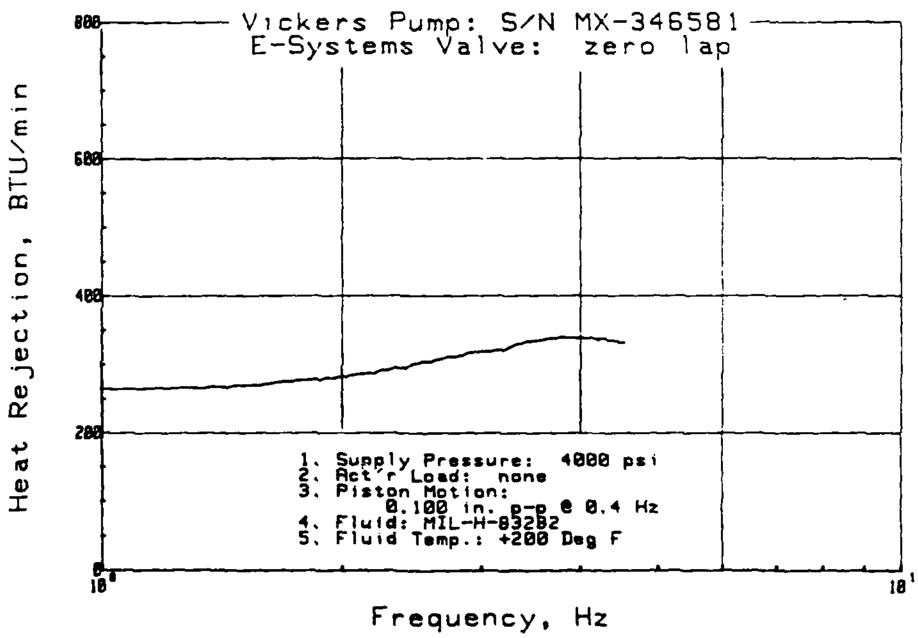


Figure 39. System energy consumption, E-Systems zero lap valve

TABLE 17. Energy consumption data, LHS simulator

E-SYSTEMS VALVE	ACTUATOR LOAD	PRESSURE LEVEL, psi	*HEAT REJECTION, BTU/min		
			PUMP	ACTUATOR	SYSTEM
Zero lap	none	4000	192	37	280
		8000	414	91	555
Zero lap	5000 lb	4000	194	25	240
		8000	425	74	548
Overlapped	none	4000	204	30	255
		8000	410	71	530
Overlapped	5000 lb	4000	198	21	242
		8000	429	58	533

*Heat rejection at 2 Hz. See Appendix E.

1.3 DEMONSTRATION PHASE SUMMARY

1.3.1 Control Valves

Evaluation of the Bendix valves was stopped when a failure occurred during dynamic testing. The failure was the result of design and was not caused by a technological problem or related to the DDV concept. Performance of the E-Systems valves was excellent during all the hardware demonstration tests. The zero lap and 0.002 in. overlap valves provided test data that clearly showed the effects of overlap on valve performance.

Operating pressure level had the following general effects on valve performance: (8000 psi data used as the basis for comparison)

	Performance at 4000 psi
Flow Gain	
Zero lap valve	0.66
Overlapped valve	0.68
Pressure Gain	
Zero lap valve	0.64
Overlapped valve	0.51
Internal Leakage	
Zero lap valve	0.74
Overlapped valve	0.34

Valve overlap had the following general effects on performance: (zero lap data used as the basis of comparison)

	Performance with 0.002 in. overlap
Flow Gain	
4000 psi	1.00
8000 psi	.99
Pressure Gain	
4000 psi	.22
8000 psi	.27
Internal Leakage	
4000 psi	.08
8000 psi	.17

As expected, pressure gain and internal leakage were affected by overlap. The reduction in internal leakage due to overlap agreed well with estimates made in Section 2.4.5 of Volume I.

1.3.2 Servo Actuator

Valve overlap had little effect on actuator response to step inputs. Actuator frequency response was affected by operating pressure level and overlap. The break frequency (-3 db point) that occurred with operation at 8000 psi was about 25% lower at 4000 psi. Fortunately, the response requirements for operation at 4000 psi are lower than the requirements for 8000 psi. Valve overlap (0.002 in.) decreased the break frequency an average of 28%. Methods to compensate for this reduction are discussed in Section 2.4.5 of Volume I.

The break frequency that occurred when the actuator was force loaded was less than one-half of the break frequency when the actuator was mass loaded. Mass loading was relatively small during the force load test. (This would tend to increase the break frequency.) The principal cause for the difference was believed to be due to the high spring rate of the mass load fixture structure compared to the relatively low spring rate of the load fixture structure in the LHS simulator.

The pressure level switching tests produced excellent results:

- o Actuator piston position disturbances were negligible
- o Pressure transients were minor.
- o The time required for the 4000 psi to 8000 psi switch-over was somewhat higher than desired but considered acceptable. *(The 8000 psi to 4000 psi switch-over time is not critical.)*

The energy consumption tests conducted on the pump/servo actuator system disclosed that pump heat rejection accounted for 85 to 90% of the total losses. Valve overlap reduced actuator losses from 15% (with zero lap) of total system losses to 10% (with 0.002 in. overlap).

The test system had one control valve. A realistic assessment of the distribution of energy losses should involve the number of servo valves employed in a typical flight control system. The test pump, rated for 10 gpm at 7800 psi, was designed for use in an A-7E aircraft, reference 1. Rated flow of the E-Systems test valve is 3.5 gpm; this could be considered as an average size valve on the A-7E. The A-7E has 8 primary flight control dual system servo actuators. Using this information as the basis for a hypothetical system, the distribution of energy losses would be:

		Percent of Total Losses	
Valve Configuration	System Pressure	Pump	8 Control Valves
Zero lap	4000	48%	52%
	8000	41%	59%
Overlapped	4000	57%	43%
	8000	50%	50%

Pump losses now account for roughly one-half of the total losses. The use of 0.002 in. of overlap reduces valve losses about 18%; this agrees well with estimates made in Section 2.4.5 of Volume I.

1.3.3 Dual Pressure Pump

The modification to convert the test pump from a single to a dual pressure level pump was successful. Pump operation at 8000 psi was approximately the same as that observed during tests in prior programs, reference 1: pressure ripple, transient response, and performance were within design requirements except for heat rejection. Operation at 4000 psi was found to be completely satisfactory. Pressure level switching produced only minor pressure transients and the time required to switch from 4000 psi to 8000 psi was under the design goal of 0.100 sec at rated speed (switching valve operating time not included).

Dual pressure level operation in a system that has a bootstrap reservoir requires a new design consideration: pump suction pressure will vary with system operating pressure level. Three options are available to handle this situation.

1. Design the pump to operate with the suction pressure provided by the reservoir when system pressure is 4000 psi.

2. Design the reservoir to provide minimum pump suction pressure when system pressure is 4000 psi.

3. Install a check valve and an accumulator in the line to the reservoir bootstrap port to trap 8000 psi pressure and provide adequate pump suction pressure when system pressure is 4000 psi. A 2-way solenoid would be required to release the trapped 8000 psi pressure for maintenance purposes.

Options 1 and 2 would add about one pound to component weight; option 3 would add about 4 pounds to the system.

1.3.4 LHS Simulator

Valve overlap had a minor effect on actuator response to step inputs. Actuator frequency response was affected by operating pressure level and overlap. The break frequency that occurred with operation at 8000 psi was about 27% lower at 4000 psi. Valve overlap (0.002 in.) reduced the break frequency an average of 32%. Methods to compensate for this decrease are discussed in Section 2.4.5 of Volume I.

Pressure level switching tests conducted on the servo actuator and system produced excellent results. Pressure transients were minor. Switching time (valve operating time + pressure level switching time) averaged 0.180 sec for the 4000 psi to 8000 psi switch. This was considered acceptable.

System pressure dynamics were excellent. Operation of the dual pressure pump did not introduce spurious frequencies or undesirable pressure pulsations.

Energy consumption tests conducted on the LHS simulator showed that losses at 4000 psi averaged 47% of those that occurred when operating at 8000 psi. This clearly demonstrated the advantage of switching to 4000 psi to conserve energy.

2.0 HARDWARE DEMONSTRATION PHASE CONCLUSIONS

Successful completion of the laboratory demonstration tests verified that:

- o Energy savings achieved through the use of dual pressure level systems and overlapped servo valves are substantial.

- o Hardware can be readily designed and fabricated for use in 4000 psi/8000 psi hydraulic systems.

Although the performance of the test pump was excellent, heat rejection was higher than desired. Pump heat rejection is an area where basic changes in pump design could have a significant impact in reducing system energy consumption. The nybrid pump discussed in Volume I is one approach toward remedying this condition.

As expected, overlapped valves offer important energy savings but at the price of impaired actuator performance. This situation can be helped by the use of an "intelligent" valve amplifier as discussed in Volume I. However, this is an area that needs additional study and verification testing.

Results of the 8000 psi/4000 psi pressure level switching tests conducted on the full scale dual system LHS simulator were excellent. No adverse operating conditions were found. The possibility of using a pressure level lower than 4000 psi, such as 3000 psi, should be considered for achieving additional energy savings.

3.0 RECOMMENDATIONS

The study conducted in Volume I established that the top four candidates for saving energy in aircraft hydraulic systems were:

Advanced materials
Dual-pressure systems
Pumps
Non-linear valves

Advanced materials provide the most benefits. Two areas in aircraft hydraulic systems that could provide the largest savings are transmission lines and actuators. The application of new titanium alloys and composites to these areas should be explored further, parts fabricated, and evaluation tests conducted.

The dual pressure level system was demonstrated to be a simple and effective method for reducing energy consumption. This concept should be examined to determine an optimum pressure for the lower operating pressure level, such as 3000 psi or 2000 psi.

Pumps are the major consumer of energy in aircraft hydraulic systems. Decreasing pump heat rejection can therefore have a significant impact on energy savings. Basic changes in the conventional aircraft pump design are needed. Innovative concepts such as the hybrid check valve pump should be pursued further.

Overlap in control valves was demonstrated to provide substantial energy savings -- but at a cost in actuator performance. Methods to alleviate this impairment have been proposed, but their potential remains to be proven. Additional study and hardware demonstrations are recommended.

Tests on flow augmentors were planned to be conducted in this program if an NAAO-Columbus IR&D study progressed sufficiently. Although good progress was made, the study was halted because of funding limitations. The application of flow augmentation to flight control actuators has good potential for conserving energy and further work in this area is warranted.

REFERENCES

Reference No.	
1	Bickel, William N. and Haning, Robert K., Fabrication and Testing of Lightweight Hydraulic System Simulator Hardware, NADC-79024-60, Rockwell International Corporation, Contract N62269-80-C-0261, January 1986, Unclassified. AD A169 884
2	Demarchi, Joseph N., Dynamic Response Test of Very High Pressure Fluid Power Systems, NR70H-533, North American Rockwell Corporation, Columbus Division, Contract N00156-70-C-1152, December 1970, Unclassified. AD 891 214L
3	LHS-8810A, Pumps, Hydraulic, Variable Delivery, 8000 psi, General Specification For, dated 15 August 1985

LIST OF ABBREVIATIONS AND SYMBOLS

A/D	analog-to-digital
AFCS	automatic flight control system
BTU/min.	British thermal units per minute
C1, C2	cylinder port No. 1, cylinder port No. 2
CPU	central processing unit
CTFE	chlorotrifluorethylene
db	decibel
DC	direct current
DDV	direct drive valve
deg	degree (angle or temperature)
dia.	diameter
EDU	electronic drive unit
F	frequency
^{o}F	degrees Fahrenheit
FC-1	flight control system No. 1
FC-2	flight control system No. 2
F.S.	full scale
gpm	gallons per minute
hp	horsepower
hr.	hour
HR	heat rejection
Hz	Hertz (cycles per second)
in.	inch
in^3	cubic inches
K	kilo (1000)
lb	pound
lb-in	pound-inches (torque)
LVDT	linear variable differential transformer
M/N	model number
min.	minute (time)

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)

MTBF	mean-time-between-failures
N	pump speed
NAAO	North American Aircraft Operations
No.	number
O.D.	outside diameter
P	pressure
ΔP	differential pressure
PC1	pressure in cylinder port No. 1
PC2	pressure in cylinder port No. 2
p-p	peak-to-peak
P/N	part number
psi	pounds per square inch
psig	pounds per square inch gage pressure
Pt.	point
PWM	pulse width modulation
Q	flow
R	return
RH	right hand
RMS	root-mean-square
rpm	revolutions per minute
RVDT	rotary variable differential transformer
sec	second (time)
S/N	serial number
sys.	system
T	torque
V	volt
UHT	unit horizontal tail

APPENDIX A

HARDWARE PROCUREMENT DOCUMENTS

Contents	Page
Direct drive valve specification	92
Dual pressure pump statement of work	106

PREPARED BY		NUMBER
R.V. HUPP	Rockwell International Corporation North American Aircraft Operations	HC284-5054
APPROVALS		TYPE PROCUREMENT
B.C. Holland	**SPECIFICATION**	DATE 24 NOVEMBER 1986
		SUPERSEDES SPEC. DATED:
	FSCM NO. 89372	REV. LTR. NEW — PAGE 1 of 14

TITLE

VALVE, DIRECT DRIVE, ELECTROHYDRAULIC,
SERVO CONTROL, 8000 PSI, ENERGY EFFICIENT

BASIC SPECIFICATION

Rockwell International
North American Aircraft Operations

FSCM NO. 89372

HC284-5054
24 November 1986

1.0 PURPOSE

This document defines hardware for use in a laboratory evaluation and demonstration of energy saving techniques applied to aircraft flow control valves. This hardware will be utilized in support of the Power Efficient Hydraulic Systems contract between Rockwell International-NAAO and the Naval Air Development Center in Warminster, Pennsylvania.

2.0 SCOPE

The objective of the Power Efficient Hydraulic System Program is to reduce overall hydraulic power requirements in Navy aircraft by lowering system demands and increasing component efficiencies. Internal leakage in actuator control valves produce significant power loss and is an area where energy saving techniques can be applied effectively to reduce overall energy consumption. One approach used in the subject program to improve the operating efficiency of flight control servo valves is the use of valve overlap and orifice shaping. The test concept is depicted in Figure 1.

One (1) energy efficient direct drive valve assembly with three (3) interchangeable spool/sleeves (or spool/sleeve/housing) and one (1) valve drive electronics package (optional) will be procured by this specification (See Note 2, Page 5). The valve assembly will be mated with an actuator via a manifold furnished by Rockwell.

Since the servo valve will be used for laboratory testing only, considerable latitude will be permitted in deviating from the requirements of this specification, providing the basic goals of the test program can be met with the proposed design. Cost should be an important consideration in concept and design selection.

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HC284-5054
24 November 1986

3.0 REQUIREMENTS

The following general requirements are for a direct drive single stage electrohydraulic flow control servo valve. Requirements specified as TBD (to be determined) shall be established by the seller and stated in the proposal.

Mechanical

o Envelope	TBD
o Mechanical Interface	ARP490 type modified for pressure
o External Null Adjustment	Required
o Spool Diameter	TBD
o Spool Travel	.020 in. (minimum)
o Chip Shear Out Force	See Note 1
o Interchangeable Spool/Sleeves	3 Required (See Note 2)
o Sleeve Rigidity	

Sleeve rigidity shall be sufficient to prevent distortion (due to pressure) from affecting valve performance.

Electrical

o Valve Drive Motor Coil(s)	
+ Quantity	2, 3, or 4 coils
+ Current	.5 to 1.5 Amp/Coil
+ Resistance	TBD
+ Inductance	TBD
o Electrical Connector	TBD
o Spool Position Transducer	
+ Redundancy	TBD (compatible with No. of coils)
+ Excitation Voltage	TBD
+ Excitation Frequency	TBD

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FSCM NO. 89372

HC284-5054
24 November 1986

Hydraulic

o Number of Hydraulic Systems	1
o Operating Pressure	8000 psi
o Proof Pressure	See Note 3
o Burst Pressure	See Note 4
o Fluid	MIL-H-83282
o Internal Leakage	Figure 2
o Orifice Shape	Figure 2
o Lap	Figure 2
o Temperature Range (Design)	-40^{o} to $+275^{o}F$
o System Fluid Cleanliness	5 Micron Filtration

Performance

o Rated Conditions	
+ Valve Pressure Drop	7000 psi
+ Inlet Fluid Temperature	$+150^{o}F$
o Static Performance	
+ Rated Flow	3.6 to 5 gpm
+ Linearity	TBD
+ Symmetry	TBD
+ Hysteresis	TBD
+ Threshold	TBD
+ Flow Gain	Figure 3
o Null	
+ Lap	Figure 2
+ Null bias	5% (of rated current)
+ Null shift	2% (of rated current)
+ Null Pressure Gain	Figure 4
o Frequence Response	Figure 5

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FSCM NO. 89372

HC284-5054
24 November 1986

Electronic Unit (Optional)

- o Valve Drive Electronics (See Note 5)
  An electronic unit to provide actuation loop closure and to operate the valve drive motor shall be provided. The unit should be capable of providing closed loop performance for each of the three spool/sleeve valve designs. A single-channel unit is acceptable.
- o Actuation Performance — Figure 6
- o Actuation Reference Data — Figure 7
- o Current limiting shall be provided to protect the valve drive motor.
- o Preferred supply power — 400 Hz 115 vac or 28 VDC

NOTES:

1. The valve drive motor shall be capable of shearing a hard steel particle (130,000 psi shear ultimate) with a cross sectional area equal to 67 percent of the largest single valve orifice area with all coils operating.

2. Where interchangeable sleeves or bodies are not practical, separate valve assemblies with identical valve drive motors may be provided.

3. Proof pressure values:
   Pressure and cylinder ports: 12,000 psi.
   Return port: 8000 psi.
   In order to utilize available valve drive motor designs and minimize costs, lower return port proof pressures will be considered. (Approval required)

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FSCM NO. 89372

HC284-5054
24 November 1986

4. Burst pressure values:

   Pressure and cylinder ports: 16,000 psi

   Return port: 12,000 psi

   In order to utilize available valve drive motor designs and minimize costs, lower return burst pressures will be considered. (Approval Required)

5. Laboratory Breadboard type electronics are acceptable.

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HC284-5054
24 November 1986

4.0 ACCEPTANCE TESTING

Acceptance tests shall be conducted on each of the three (3) valve configurations by the seller prior to delivery of the hardware to Rockwell. Recommended acceptance test procedures to establish baseline performance characteristics shall be prepared by the seller and approved by Rockwell. The tests shall include as a minimum:

- Proof pressure
- Flow gain
- Pressure gain
- Internal leakage
- Frequency response

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FSCM NO. 89372

HC284-5054
24 November 1986

DEMONSTRATION HARDWARE

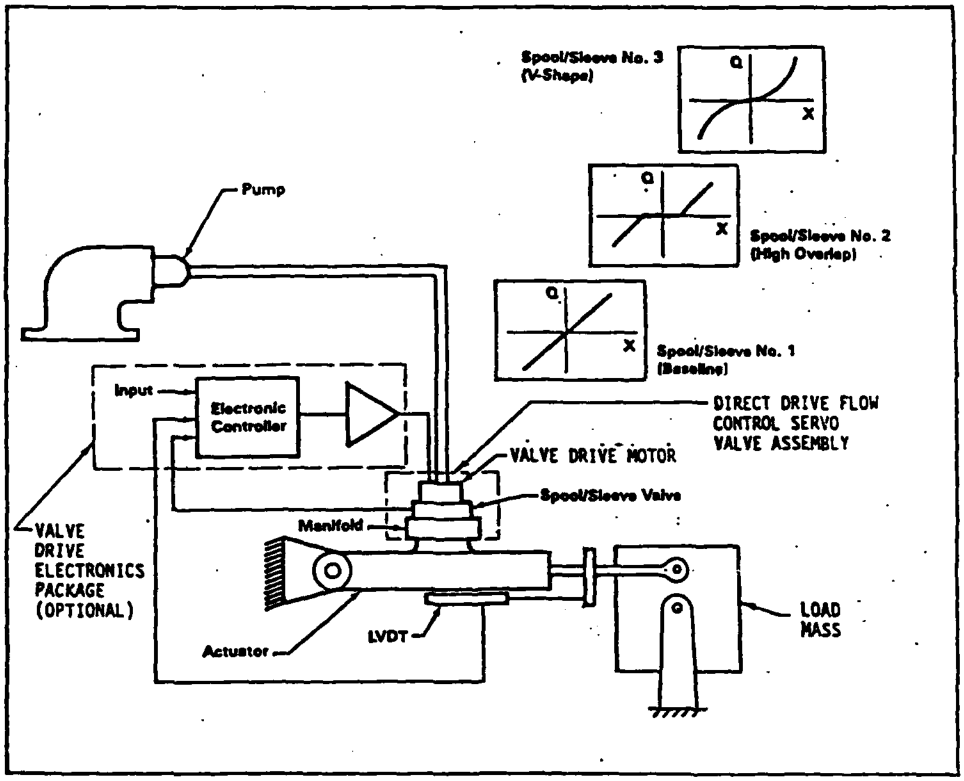

FIGURE 1

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INTERNAL LEAKAGE

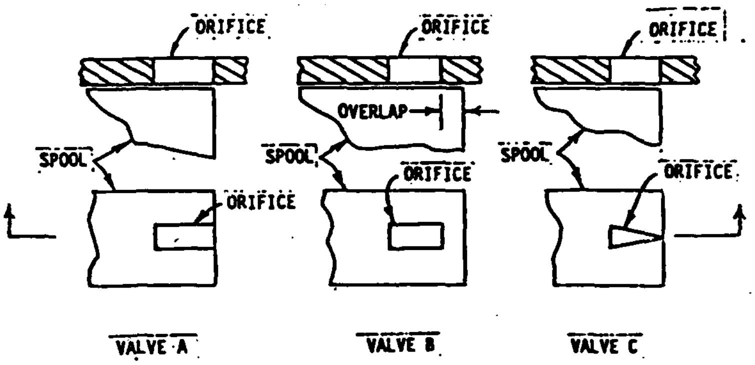
VALVE	TYPE	LEAKAGE*
A	ZERO LAP	0.1 GPM (MAX)
B	HIGH OVERLAP	TBD
C	V SHAPE	TBD

OVERLAP

VALVE	OVERLAP IN
A	0
B	.004 (MIN)
C	0

*TEMPERATURE EFFECTS TO BE ESTIMATED

ORIFICE SHAPES



(TYPICAL, VARIATIONS ACCEPTABLE)

FIGURE 2 VALVE TYPES

9

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FSCM NO. 89372

HC284-5054
24 November 1986

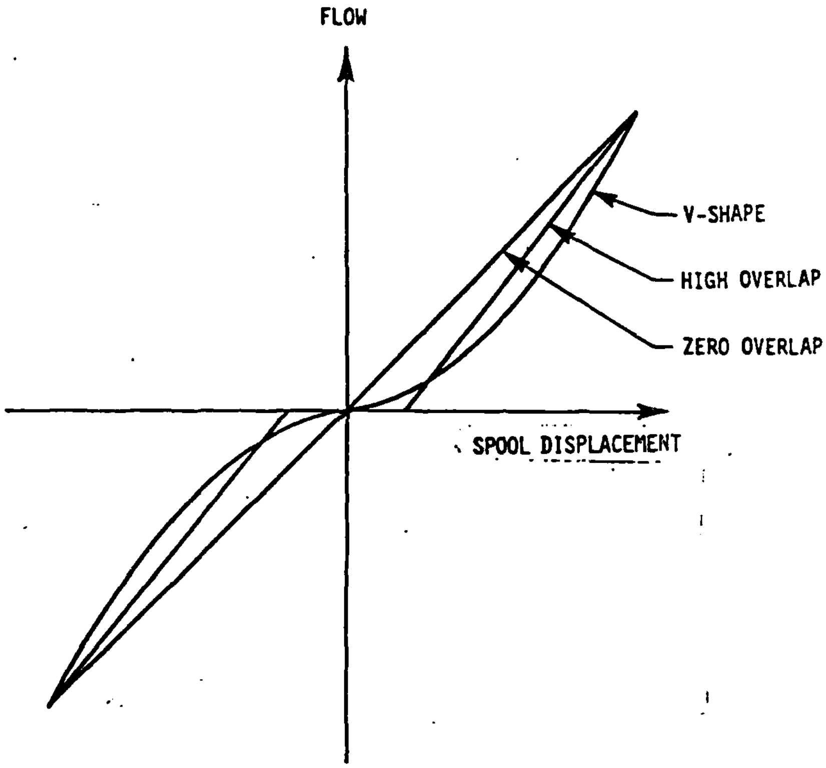

FIGURE 3 TYPICAL FLOW GAIN CHARACTERISTICS

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HC284-5054
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NULL PRESSURE GAIN (K_p), PSI/IN

VALVE	MINIMUM 3000 PSI	MINIMUM 8000 PSI
A	2×10^6	4×10^6
B	TBD	TBD
C	TBD	TBD

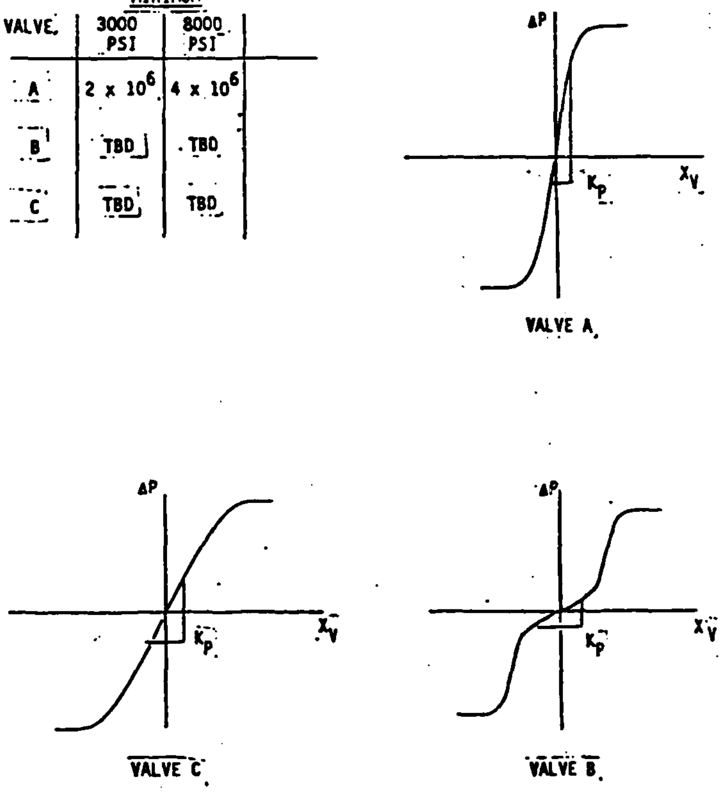

FIGURE 4 TYPICAL PRESSURE GAIN CHARACTERISTICS

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HC284-5054
24 November 1986

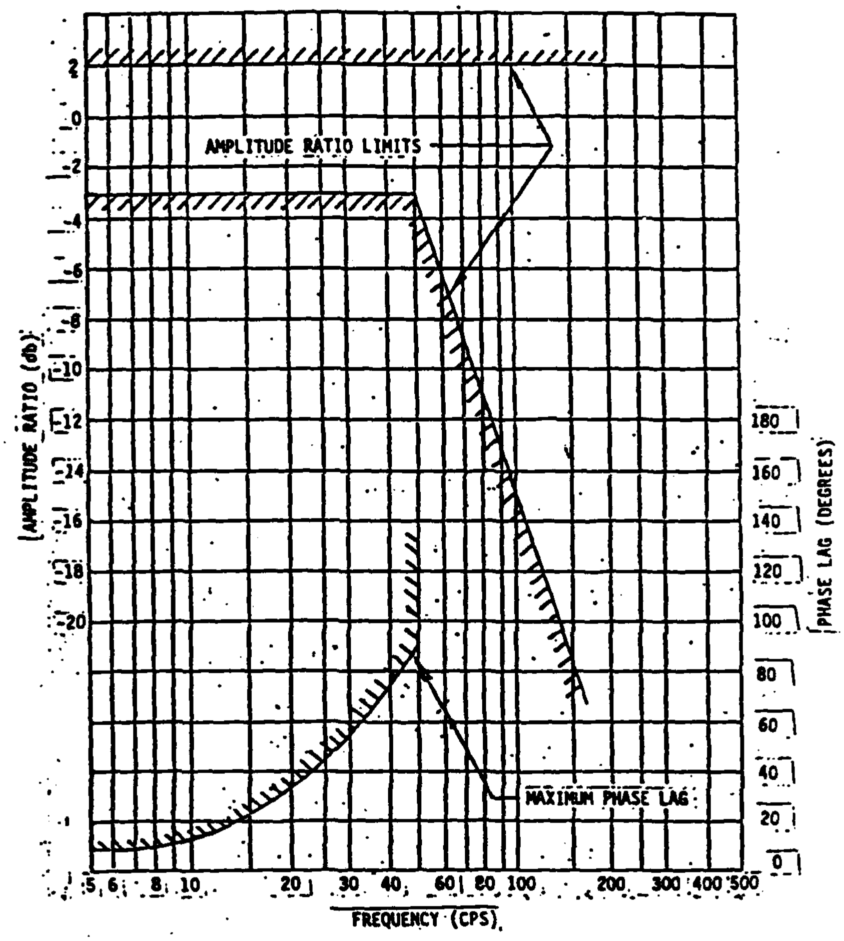

FIGURE 5 VALVE FREQUENCY RESPONSE

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HC284-5054
24 November 1986

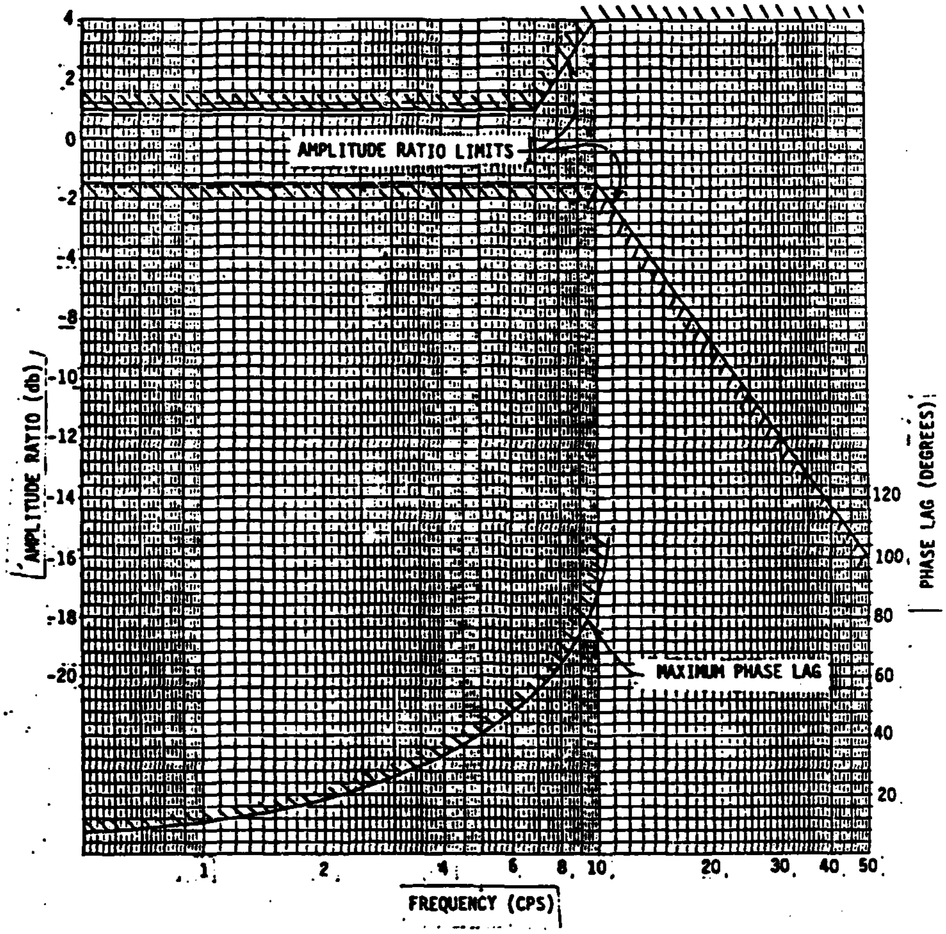

FIGURE 6 ACTUATION FREQUENCY RESPONSE (REFERENCE DATA)

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FSCM NO. 89372

HC284-5054
24 November 1986

FIGURE 7

ACTUATION REFERENCE DATA

Actuator Type	Balanced
Piston Diameter	2.368 in.
Rod Diameter	1.185 in.
Piston Area	3.301 in.^{2}
Stroke	6.58 in.
Design Pressure	8000 psi
Load Inertia	1000 in-lb-sec^2
Moment Arm	12 in.
Load Weight	2000 lbs.

Rockwell International
North American Aircraft Operations
NA-86-0081
2 April 1987
Revised: 15 May 1987

STATEMENT OF WORK

FOR

ENERGY EFFICIENT

DUAL PRESSURE PUMP

Rockwell International
North American Aircraft Operations

NA-86-0081

1.0 SCOPE

This statement of work defines the tasks required to modify a variable displacement 8000 psi hydraulic pump to demonstrate dual pressure level system operation. The pump to be modified is a Vickers Model PV3-047-2 developed for the Lightweight Hydraulic System Program. The program schedule is included as Attachment 1 hereto.

This work shall be accomplished by an aircraft hydraulic pump supplier, hereinafter referred to as the Seller, for Rockwell International Corporation, hereinafter referred to as the Buyer.

2.0 PROGRAM TASKS

2.1 Design. - The Seller shall design the modification of a Vickers Model Number PV3-047-2 pump to provide the capability for dual pressure operation at 8000 psi or 4000 psi. The modification will permit hydraulic switching from the high pressure (8000 psi) to the low pressure (4000 psi) mode of operation and from the low pressure to the high pressure mode. Control pressure shall be ported to the pump compensator mechanism using an 8000 psi, 3-port, 2-position, normally closed, pilot operated selenoid valve (Sterer P/N 15390-1). Loss of electrical power to the valve will revert the pump output to the high pressure (8000 psi) mode.

Requirements for documenting the modification design are specified on the SDRL, Attachment 2 hereto.

2.2 Performance Requirements. - The original pump design, in general, meets the performance requirements of Specification LHS-8810. The modified pump shall meet the requirements of the Specification Sheet, Attachment 3 hereto.

2.3 Fabrication/Modification. - The Seller shall fabricate the hardware to modify two (2) Vickers Model Number PV3-047-2 pumps and perform the modifications in accordance with the design developed under 2.1.

2.4 Checkout Tests. - The Seller shall conduct inspection and checkout tests on the modified pumps to verify dual mode pressure operation in accordance with the Seller-prepared Buyer-approved test procedures (See SDRL, Attachment 2). The tests shall include as a minimum:

a. Examination of Product
b. Pump run-in and dual pressure operation to total two (2) hours.
c. Determine at high and low mode pressure operation:

    (1) Efficiency
    (2) Heat Rejection
    (3) Ripple
    (4) Switching pressure transients and time.

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NA-86-0081

The Seller shall notify the Buyer at least two (2) weeks prior to the start of testing so that a Buyer representative may witness the testing, if desired.

2.4.1 Redesign and Retest. - In the event a modified pump should malfunction, the Seller shall notify the Buyer for resolution of any problem prior to proceeding with any design change or retest.

2.5 Buyer Provided Hardware. - The Buyer shall deliver to the Seller at contract go-ahead, two (2) Vickers Model Number PV3-047-2 Pumps, Serial Numbers 346581 and 346580, and one (1) Sterer Valve, P/N 15390-1, Serial Number 2, to be utilized in the performance of the effort herein.

2.6 Delivery. - The Seller shall deliver the modified pumps and valve of 2.5, following the checkout tests of paragraph 2.4, to the Buyer's facility not later than seven (7) months after contract go-ahead. The Seller shall preserve and package the items in accordance with good commercial practice in a manner that will provide protection from damage during delivery to the Buyer by common carrier.

2.7 Data and Reviews. - The Seller shall prepare and provide data to the Buyer in accordance with the attached SDRL, Attachment 2 hereto.

The Seller shall host a program review five (5) months after contract go-ahead at which time program status will be discussed.

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Attachment (3)
to NA-86-0081

PERFORMANCE REQUIREMENTS SPECIFICATION SHEET

1. BASIC INFORMATION

Pump Type:	Variable delivery, pressure compensated, axial position
Hydraulic Fluid:	MIL-H-83282
Fluid Filtration:	5 micron
Rated Speed:	5900 rpm
Operating Speed Range:	3400 to 6000 rpm
Maximum Discharge Pressure:	8000 psi
Rated Discharge Flow:	10 gpm
Rated Operating Temperature:	+200°F (Inlet Fluid)
Transient Response Time:	0.050 sec (max)
Suction Pressure:	50 psig

2. MODIFIED PUMP PERFORMANCE GOALS

	High Pressure Mode	Low Pressure Mode
Full Cut-off Pressure:	8000 ± 100 psi	4000 ± 150 psi
Maximum Full-flow Pressure:	7850 ± 150 psi	3750 ± 200 psi
Maximum Pump Ripple (peak to peak):	± 200 psi	± 250 psi
Operating Efficiency (min): (full flow)	85%	85%
Heat Rejection (max): (at 0.5 gpm discharge flow)	330 BTU/min	200 BTU/min
Mode Switching Time:	0.100 sec (max)	
Mode Switching Pressure: Transient (125 in^3 system)	9600 psi (max)	

3. DEVIATIONS

Deviations to these requirements require approval by Rockwell.

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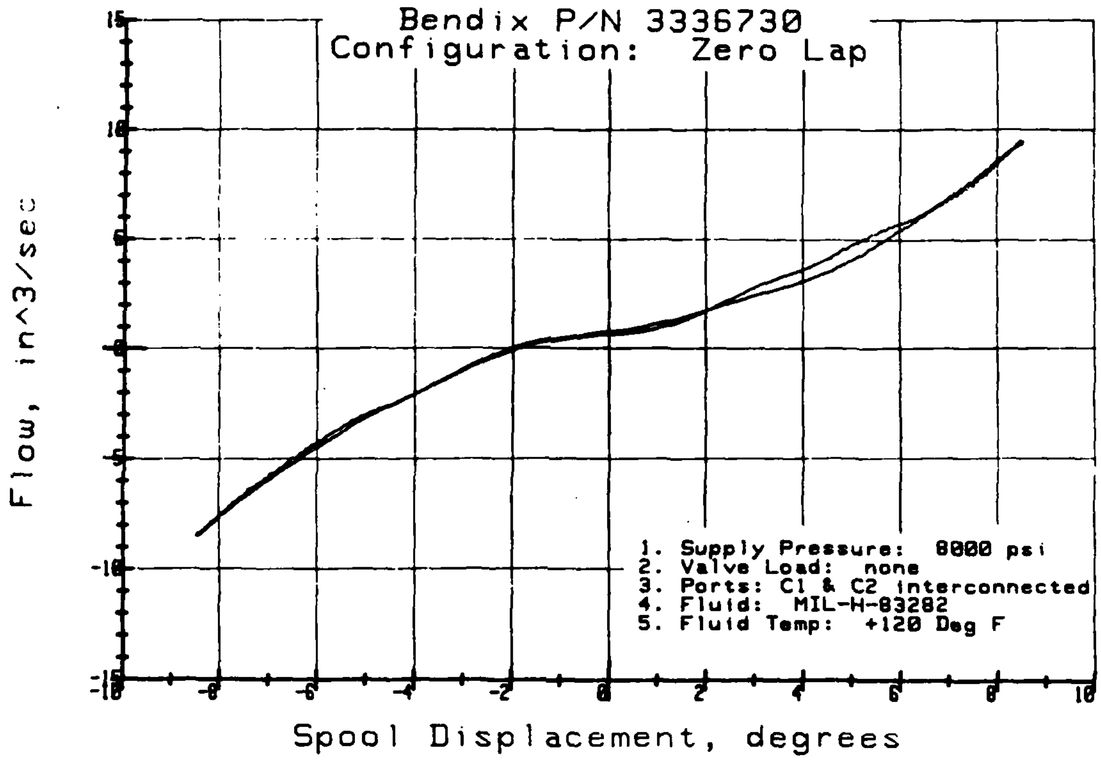
APPENDIX B

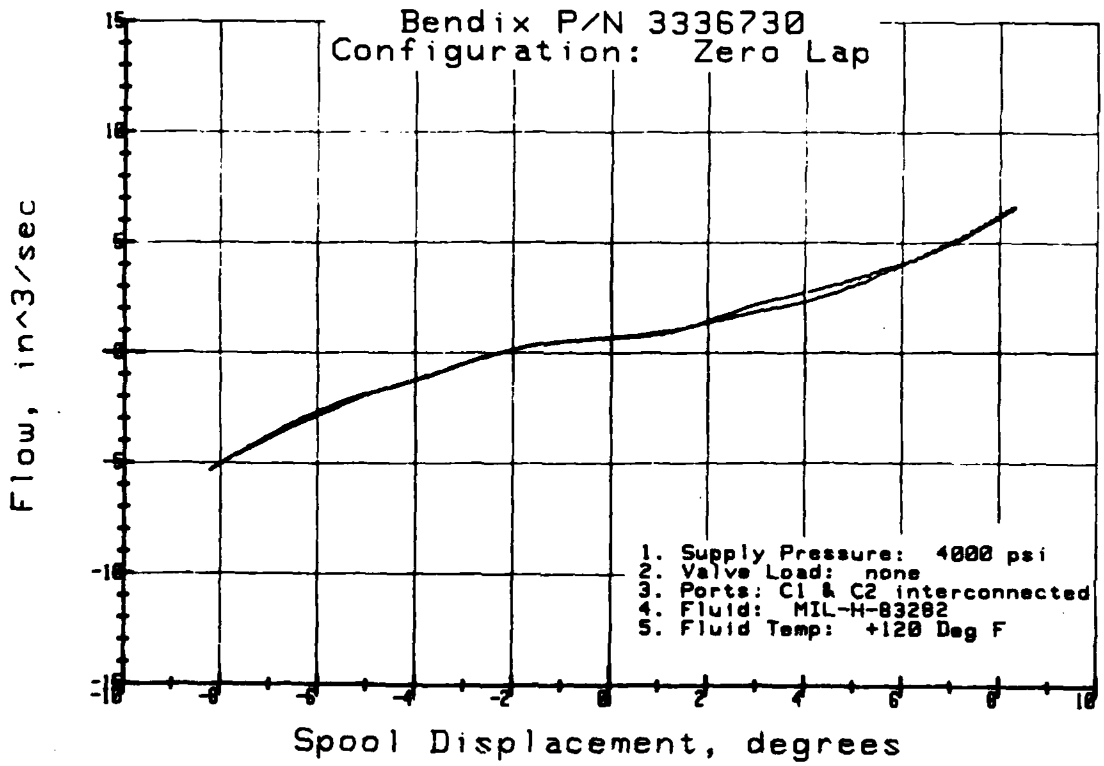
CONTROL VALVE TEST DATA

Contents | Page

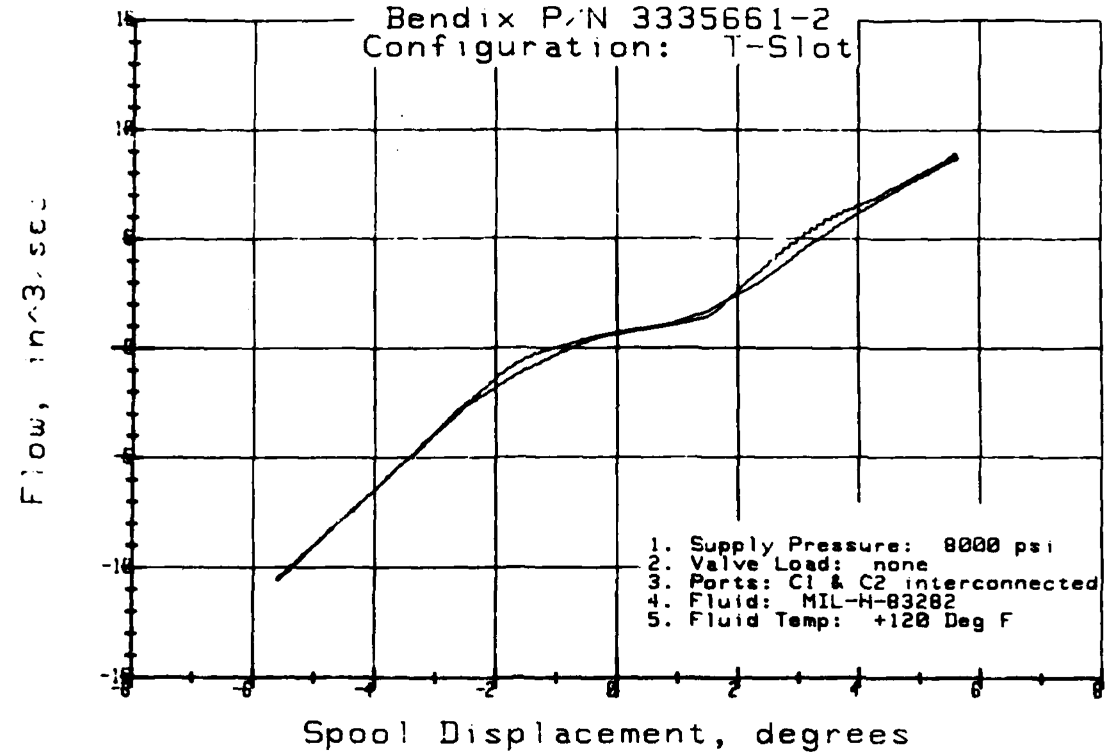
Contents	Page
Flow gain, Bendix zero lap valve	112
Flow gain, Bendix T-slot valve	113
Flow gain, E-Systems zero lap valve	114
Flow gain, E-Systems overlapped valve	115
Pressure gain, Bendix zero lap valve	116
Pressure gain, Bendix T-slot valve	117
Pressure gain, E-Systems zero lap valve	118
Pressure gain, E-Systems overlapped valve	119
Internal leakage, Bendix zero lap valve	120
Internal leakage, Bendix T-slot valve	121
Internal leakage, E-Systems zero lap valve	122
Internal leakage, E-Systems overlapped valve	123
Frequency response, Bendix zero lap valve	124
Frequency response, Bendix T-slot valve	125
Frequency response, E-Systems overlapped valve	126

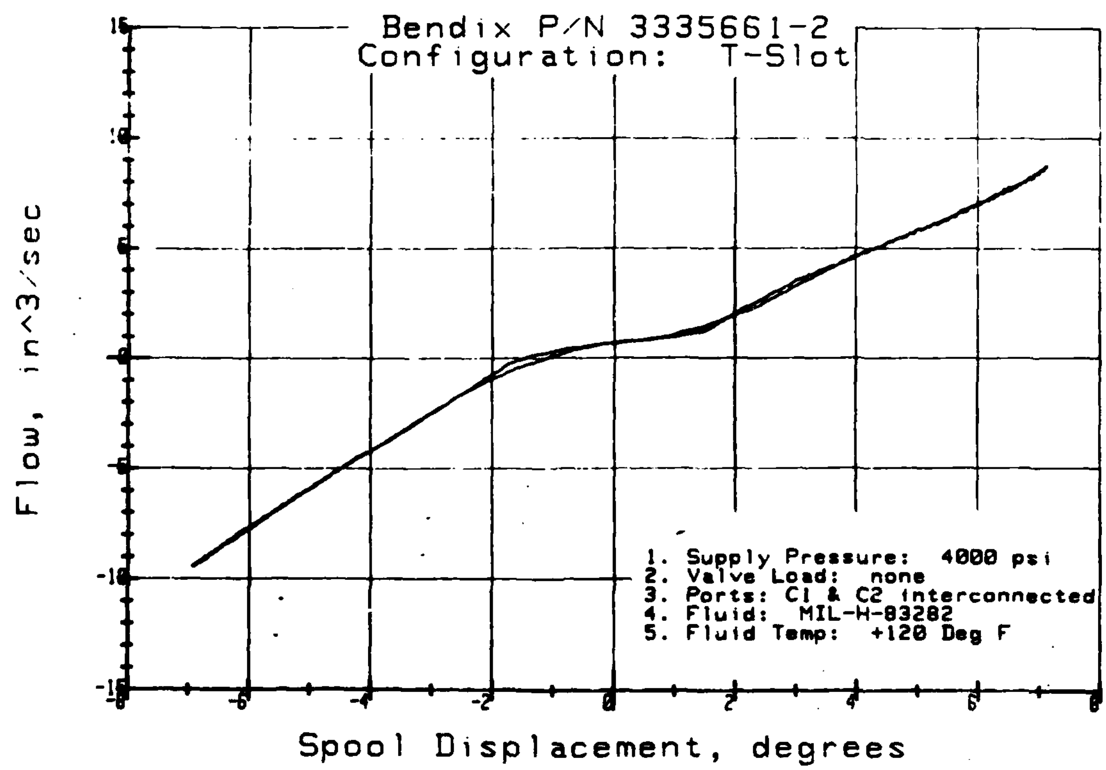
FLOW GAIN


Bendix P/N 3336730
Configuration: Zero Lap
Flow, in^3/sec
15
10
5
0
-5
-10
-15
-10
-8
-6
-4
-2
0
2
4
6
8
10
1. Supply Pressure: 8000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F
Spool Displacement, degrees

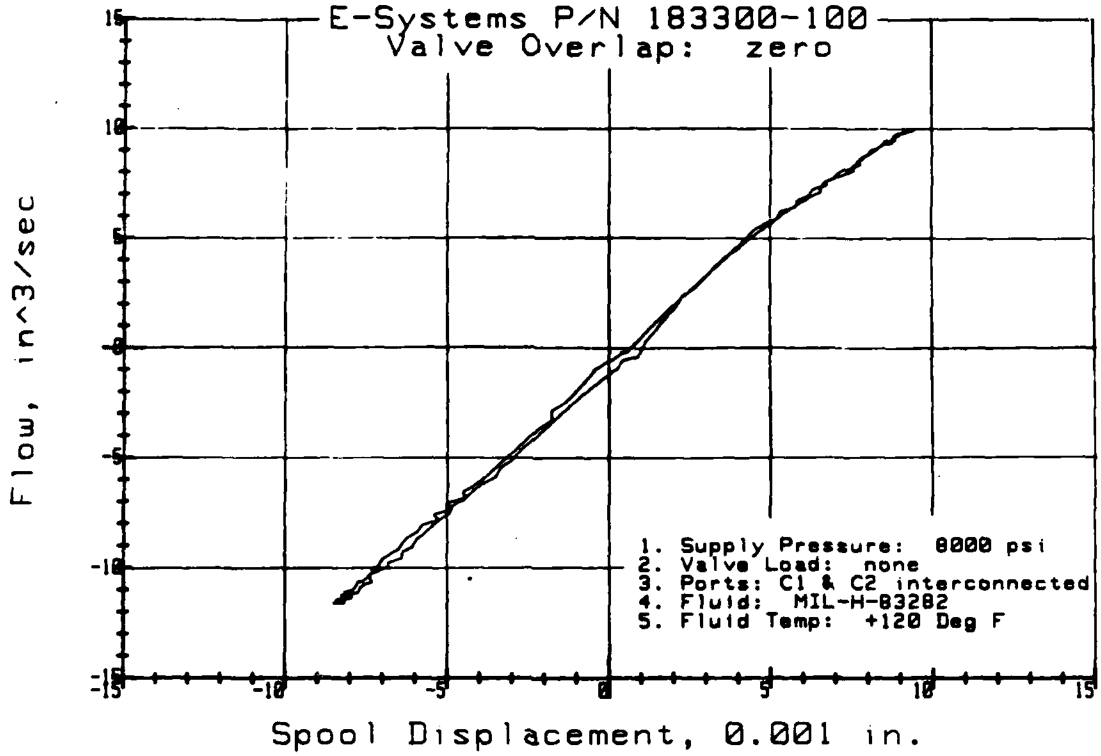

Bendix P/N 3336730
Configuration: Zero Lap
Flow, in^3/sec
15
10
5
0
-5
-10
-15
-10
-8
-6
-4
-2
0
2
4
6
8
10
1. Supply Pressure: 4000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F
Spool Displacement, degrees

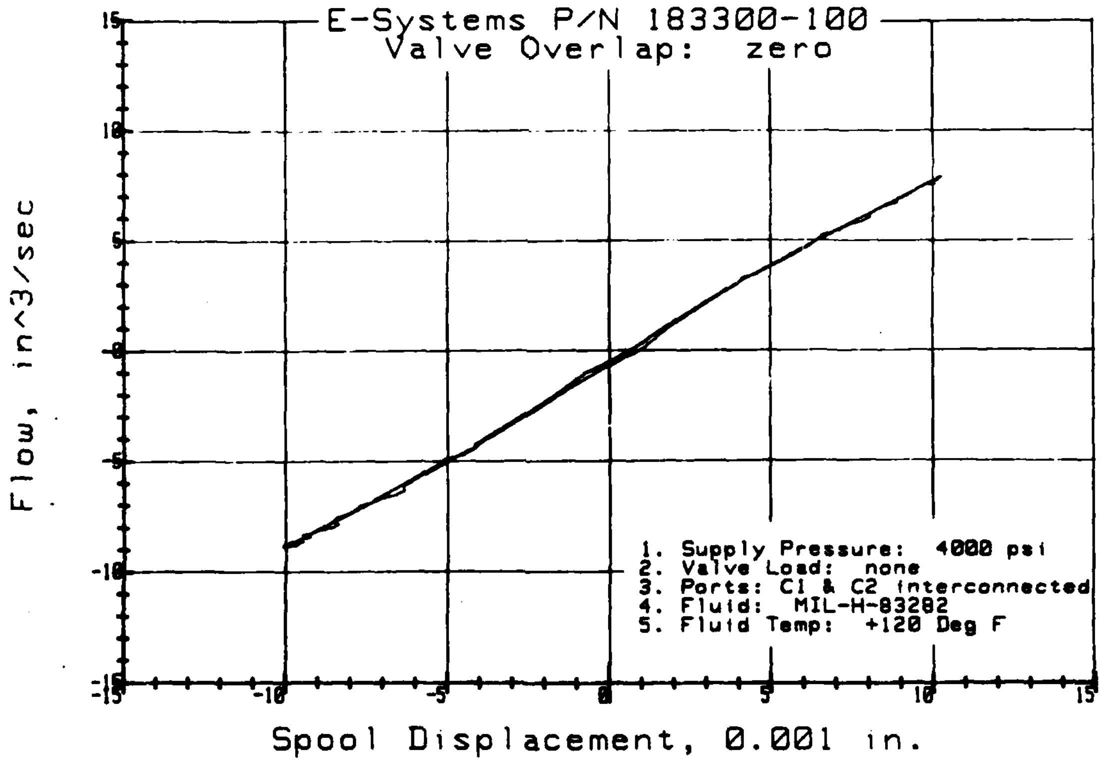
FLOW GAIN


Bendix P/N 3335661-2
Configuration: T-Slot
Flow, in^3/sec
Spool Displacement, degrees
1. Supply Pressure: 8000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

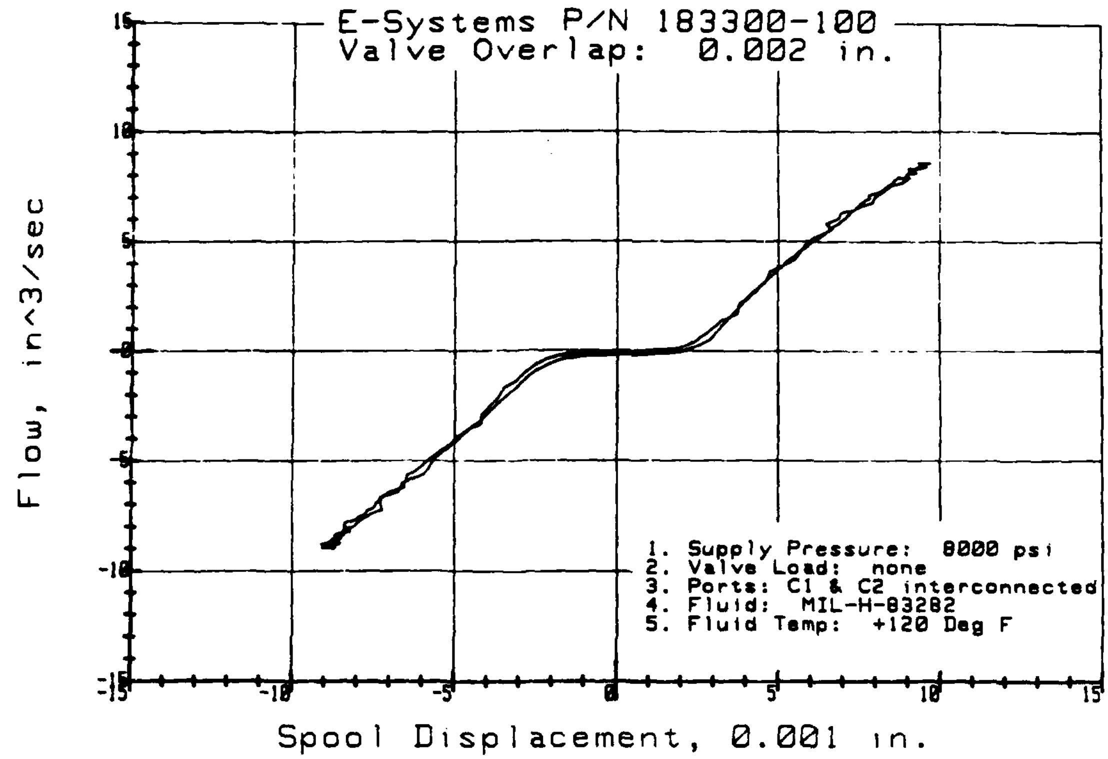

Bendix P/N 3335661-2
Configuration: T-Slot
Flow, in^3/sec
Spool Displacement, degrees
1. Supply Pressure: 4000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

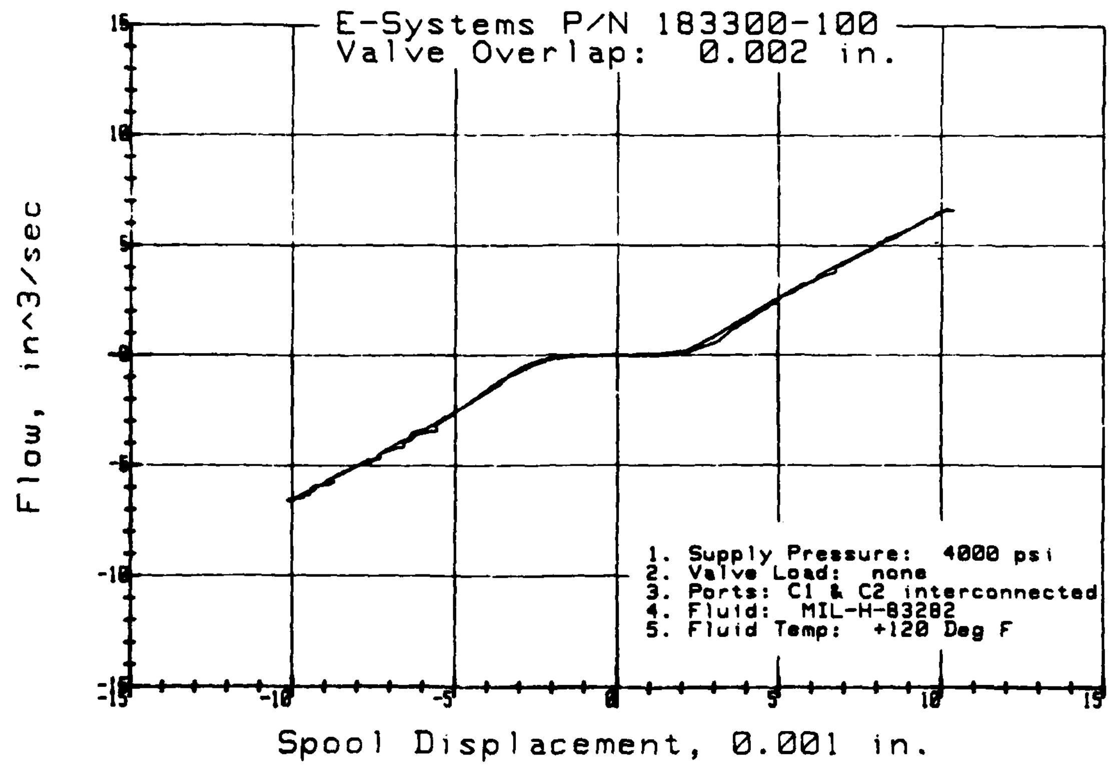
FLOW GAIN


E-Systems P/N 183300-100
Valve Overlap: zero
Flow, in^3/sec
Spool Displacement, 0.001 in.
15
10
5
0
-5
-10
-15
1. Supply Pressure: 8000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

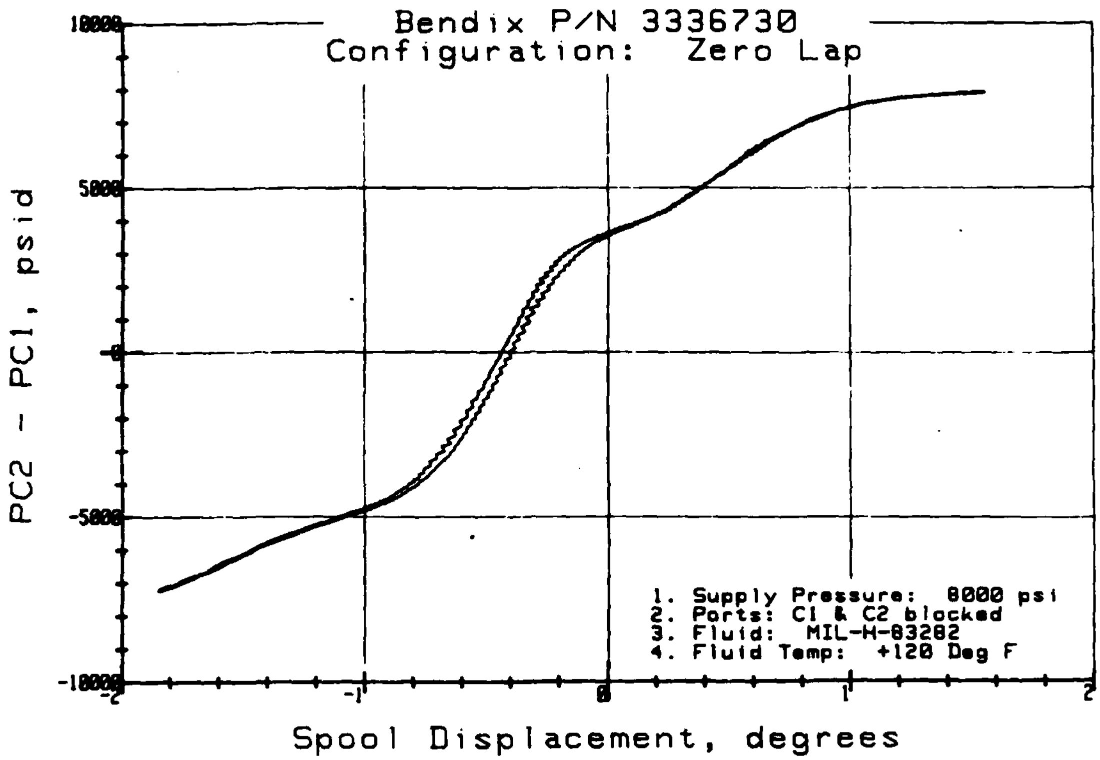

E-Systems P/N 183300-100
Valve Overlap: zero
Flow, in^3/sec
Spool Displacement, 0.001 in.
15
10
5
0
-5
-10
-15
1. Supply Pressure: 4000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

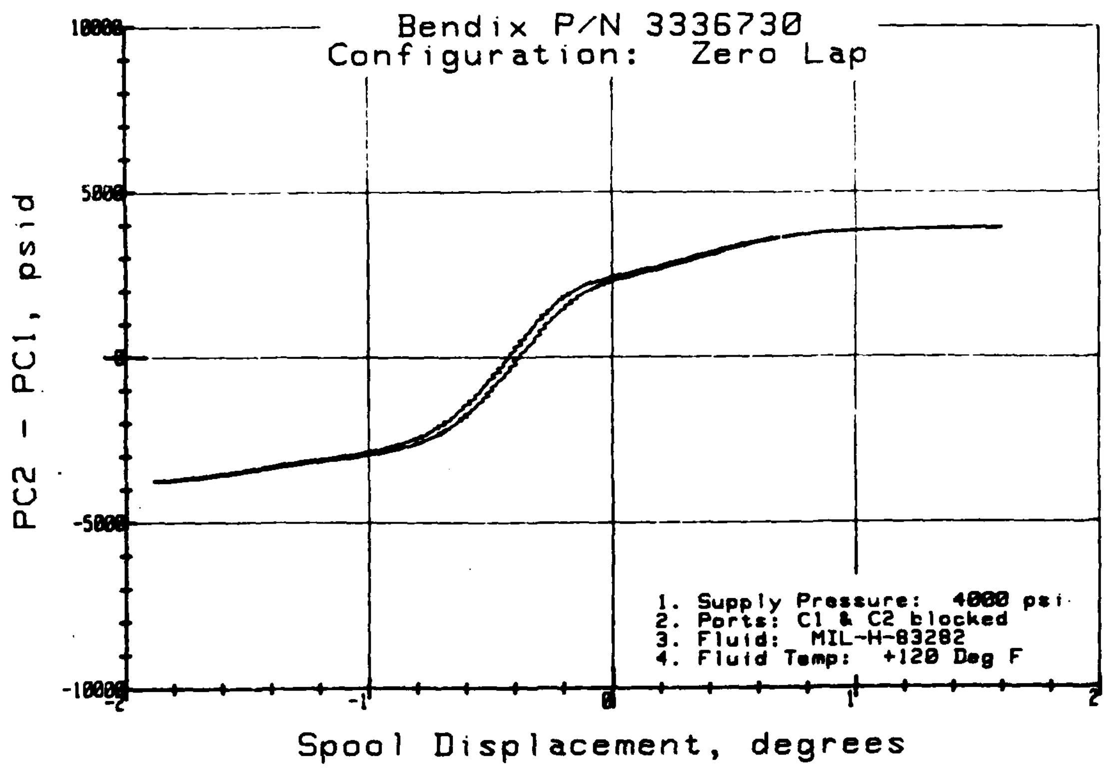
FLOW GAIN


E-Systems P/N 183300-100
Valve Overlap: 0.002 in.
Flow, in^3/sec
Spool Displacement, 0.001 in.
1. Supply Pressure: 8000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

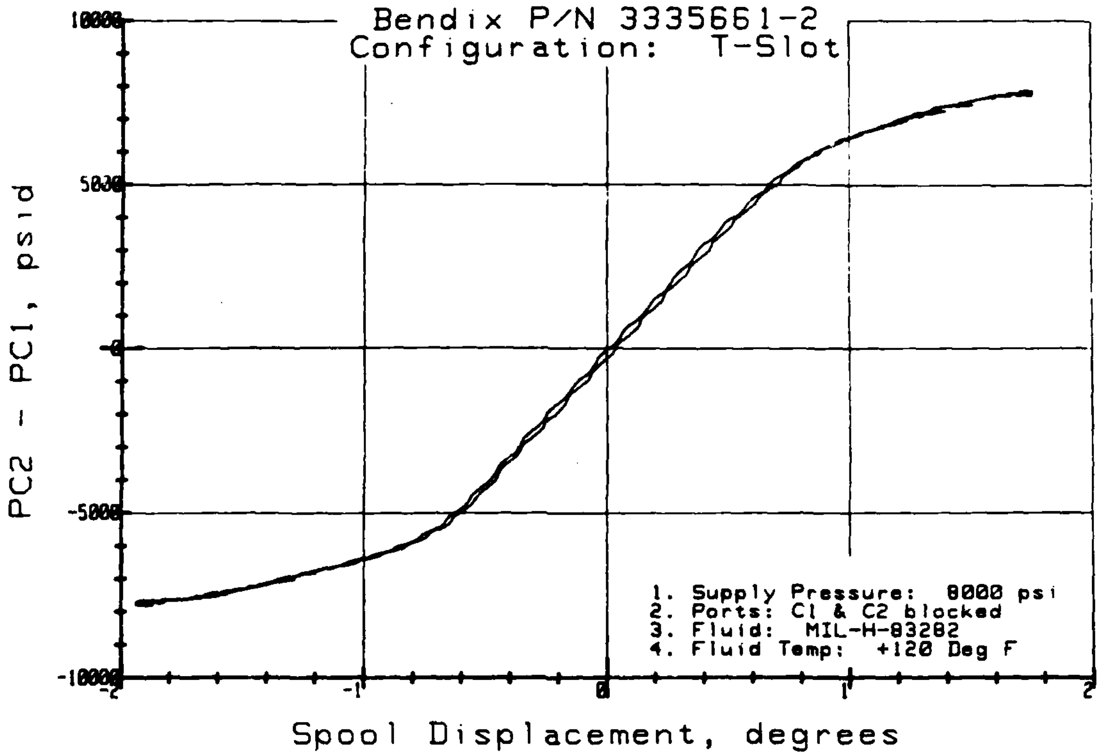

E-Systems P/N 183300-100
Valve Overlap: 0.002 in.
Flow, in^3/sec
Spool Displacement, 0.001 in.
1. Supply Pressure: 4000 psi
2. Valve Load: none
3. Ports: C1 & C2 interconnected
4. Fluid: MIL-H-83282
5. Fluid Temp: +120 Deg F

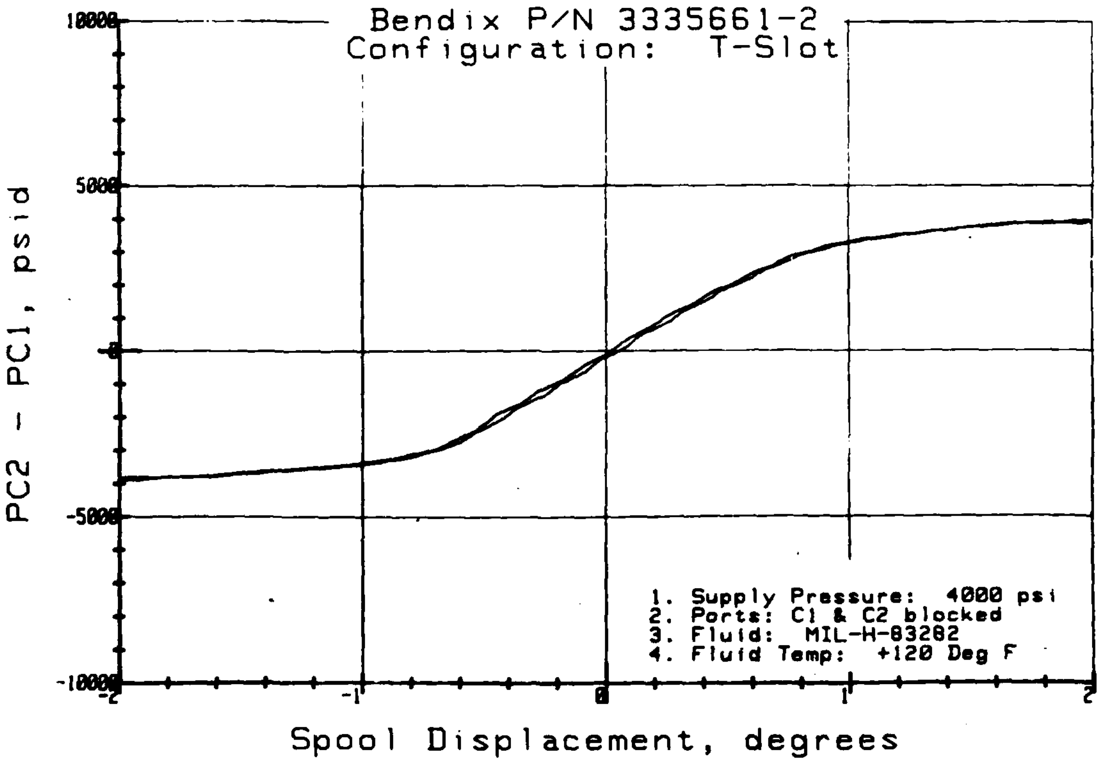
PRESSURE GAIN



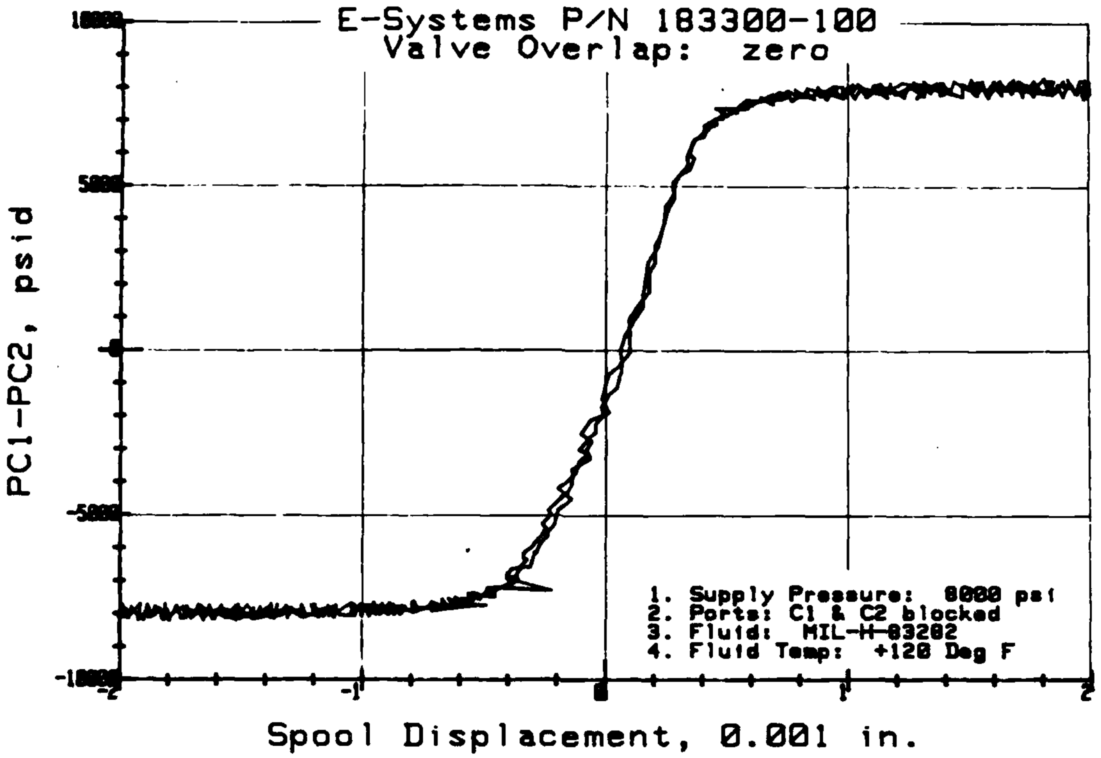


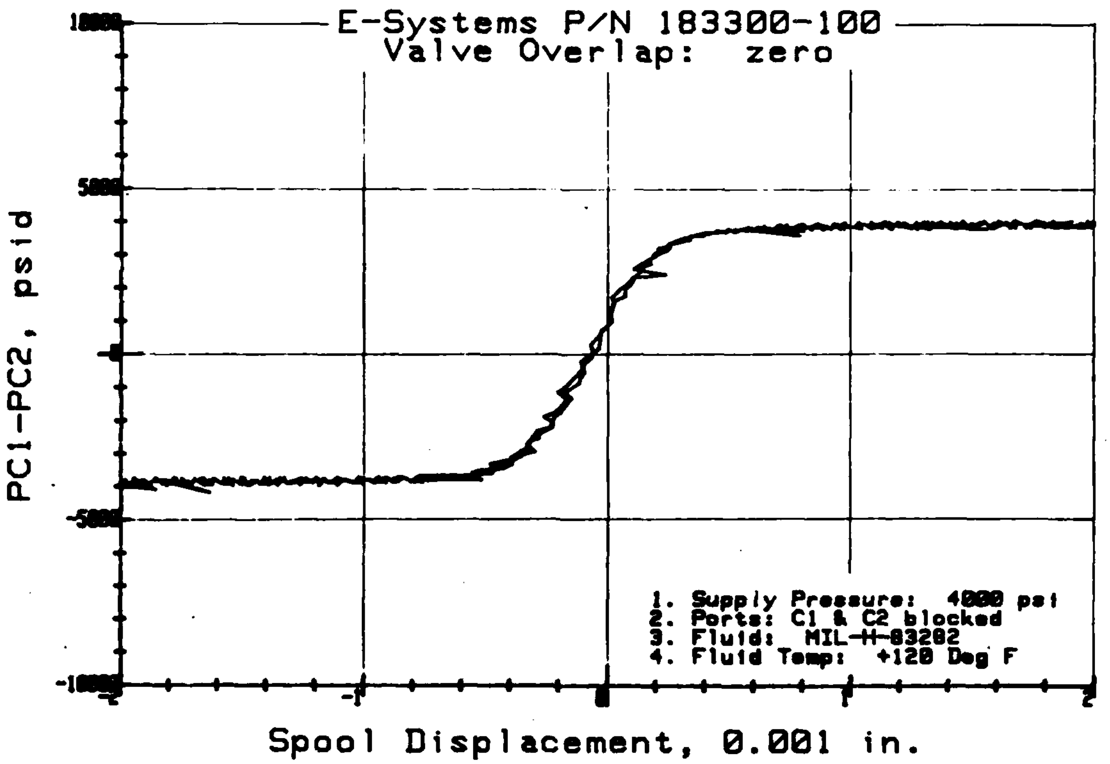
PRESSURE GAIN



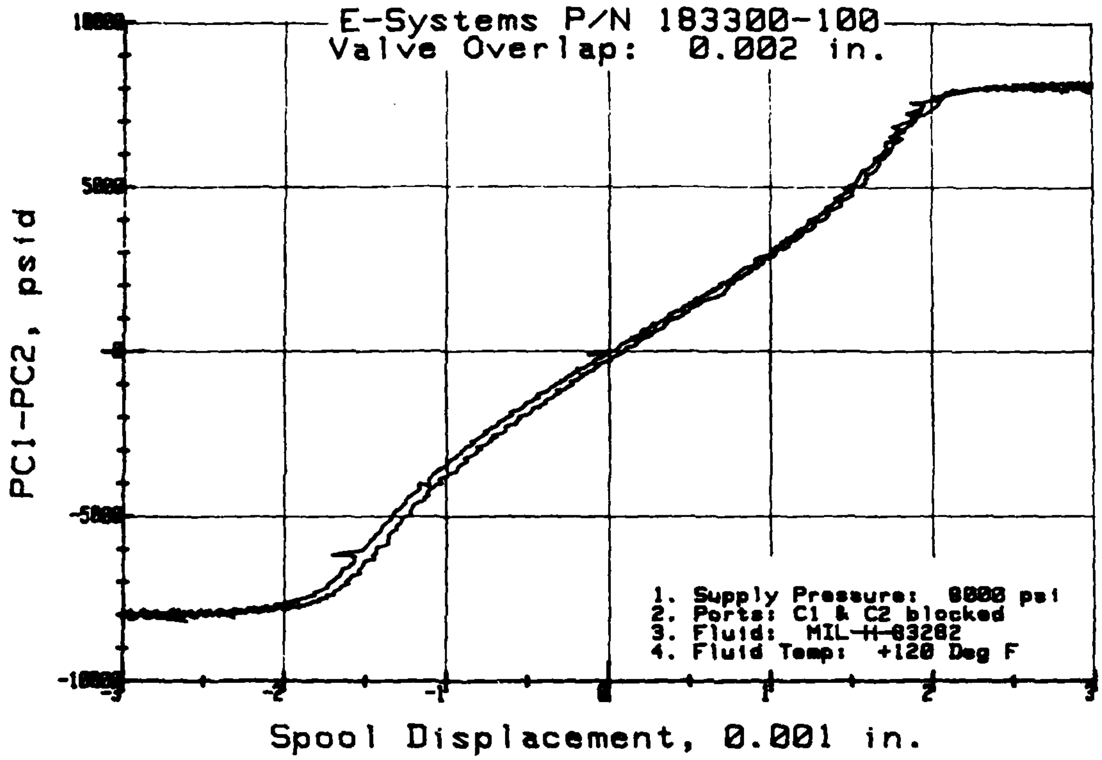


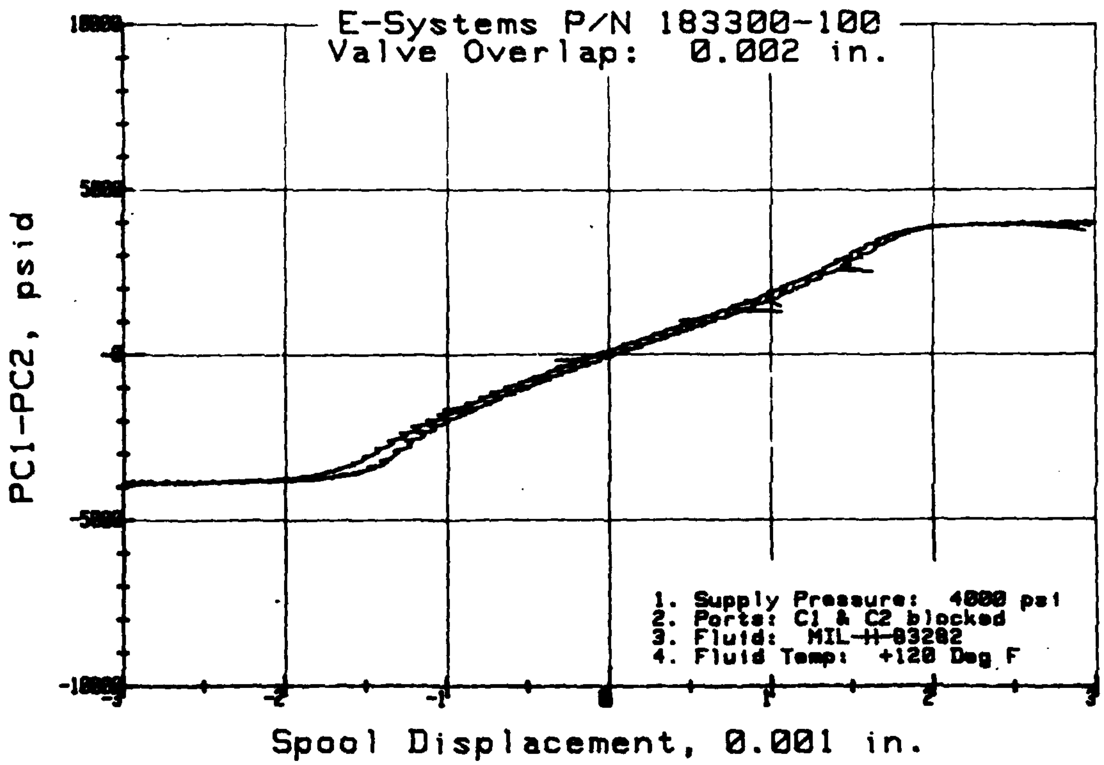
PRESSURE GAIN



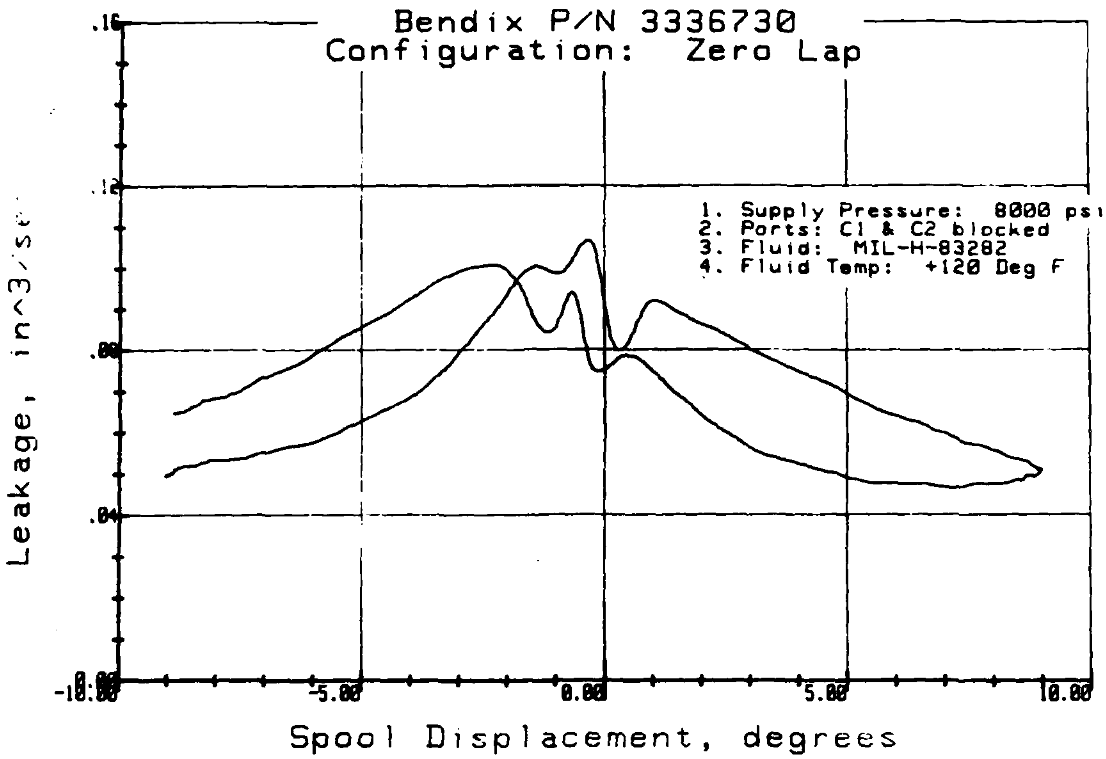


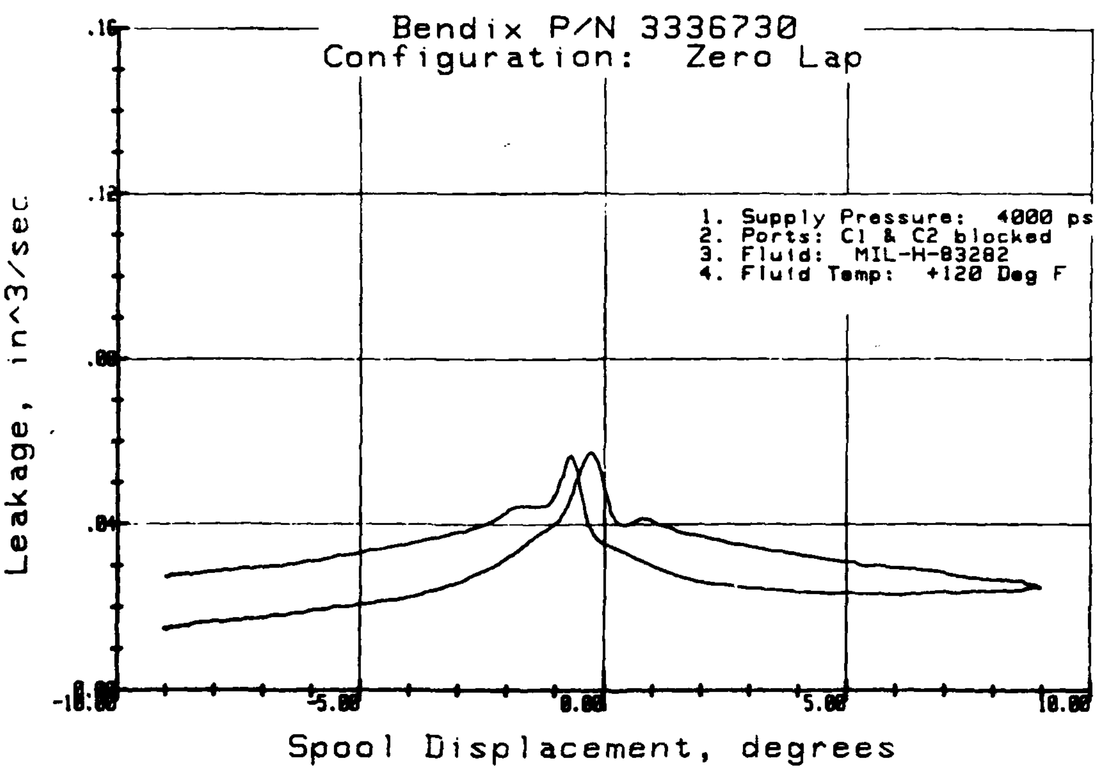
PRESSURE GAIN



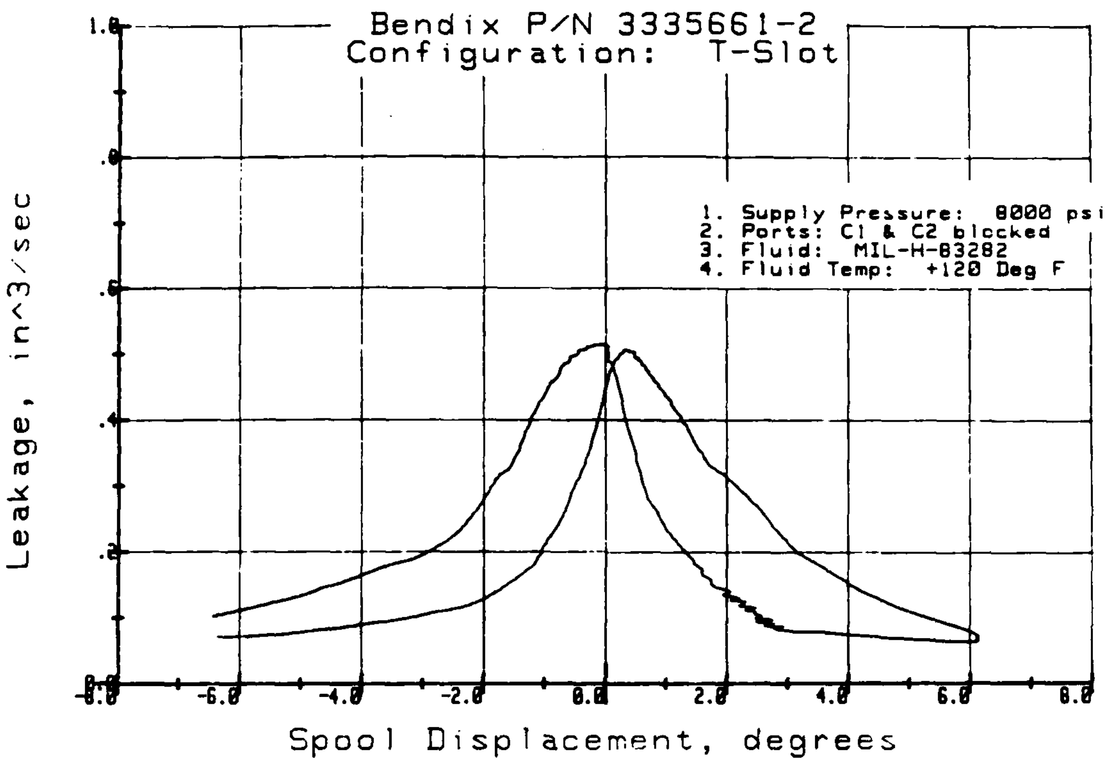


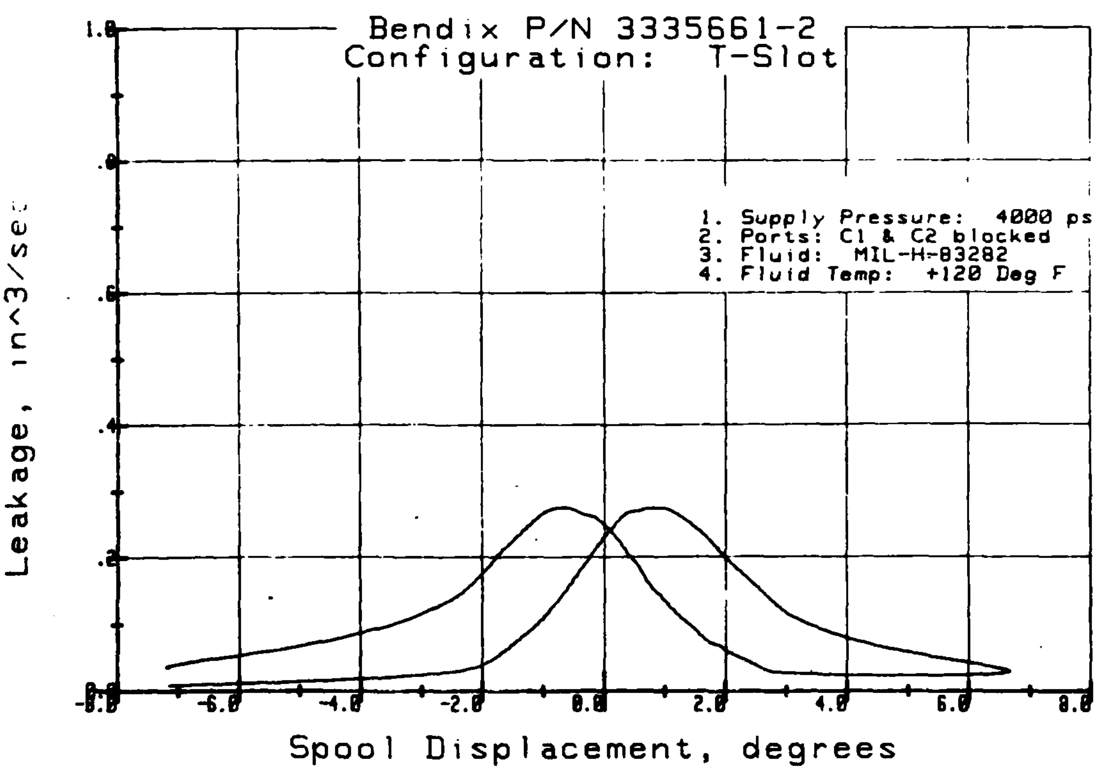
INTERNAL LEAKAGE



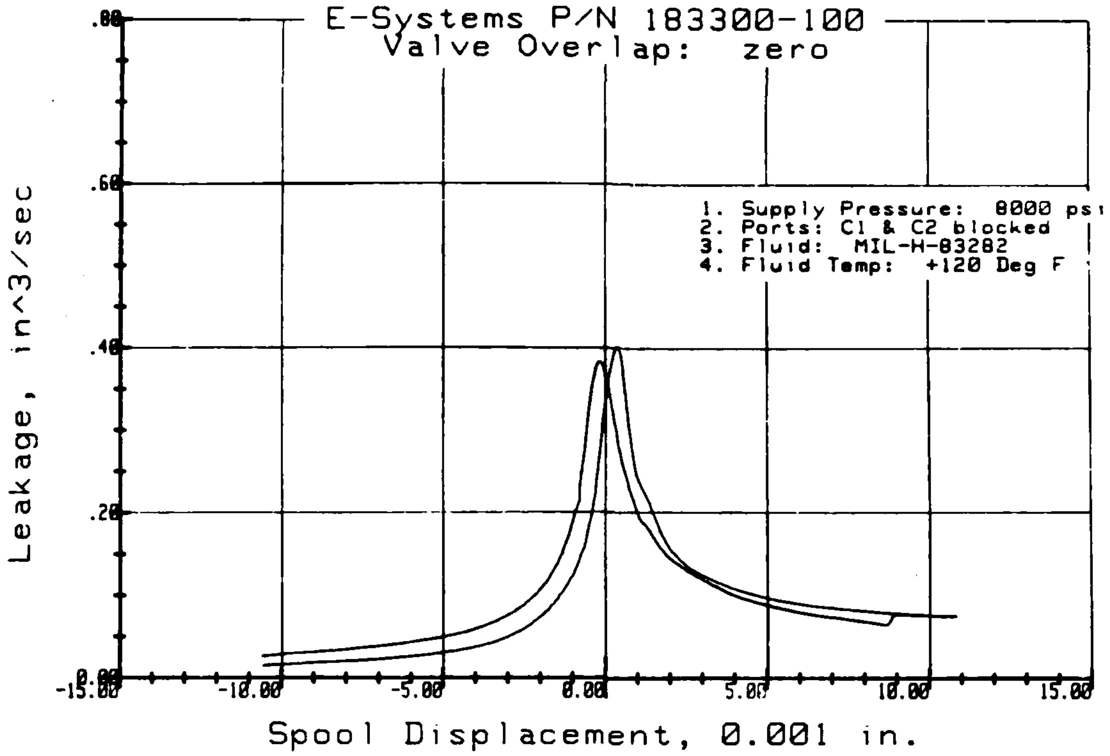


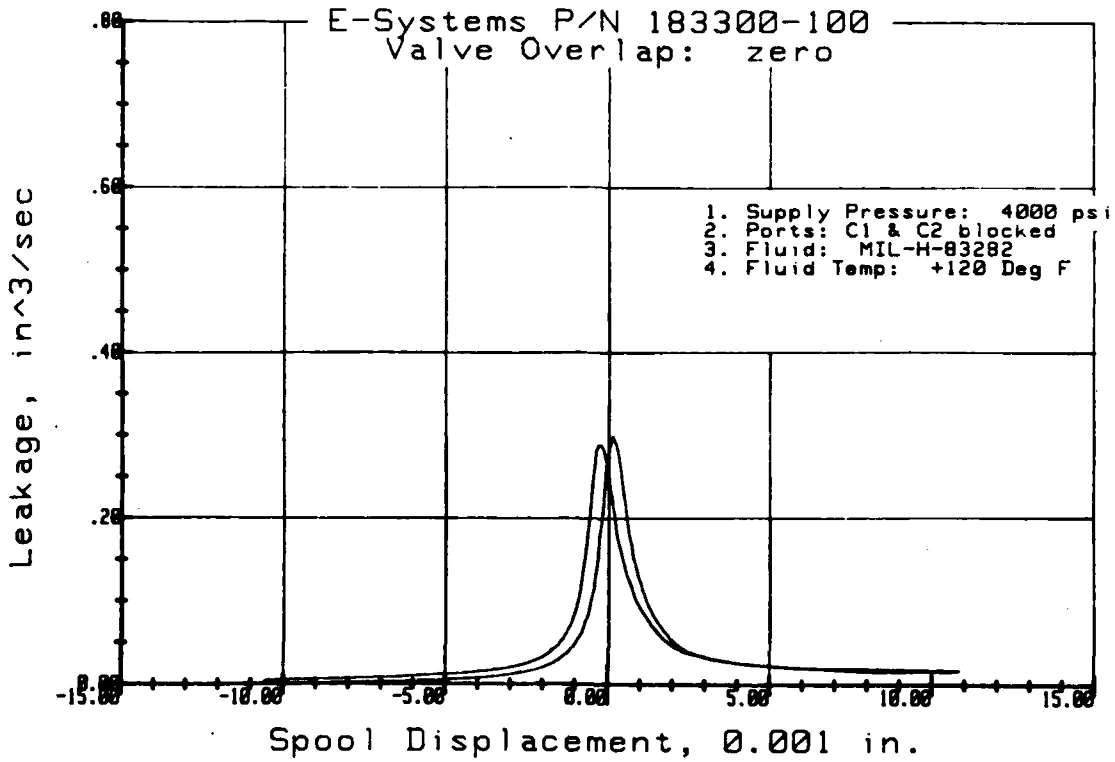
INTERNAL LEAKAGE



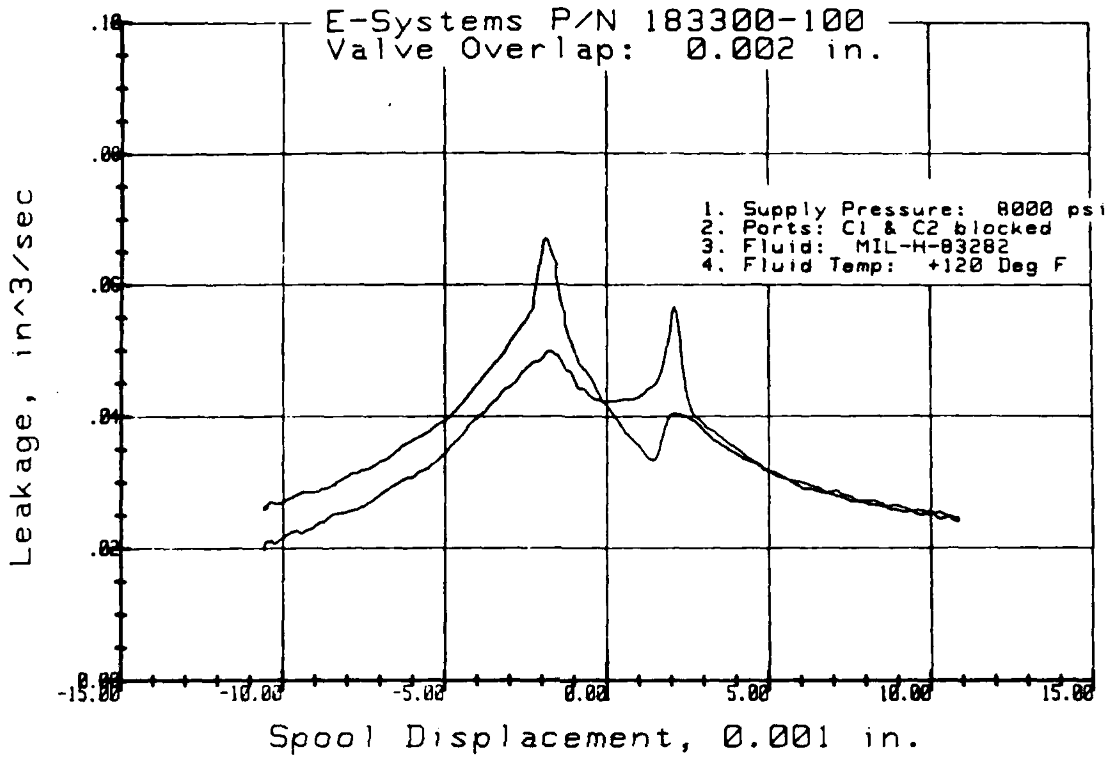


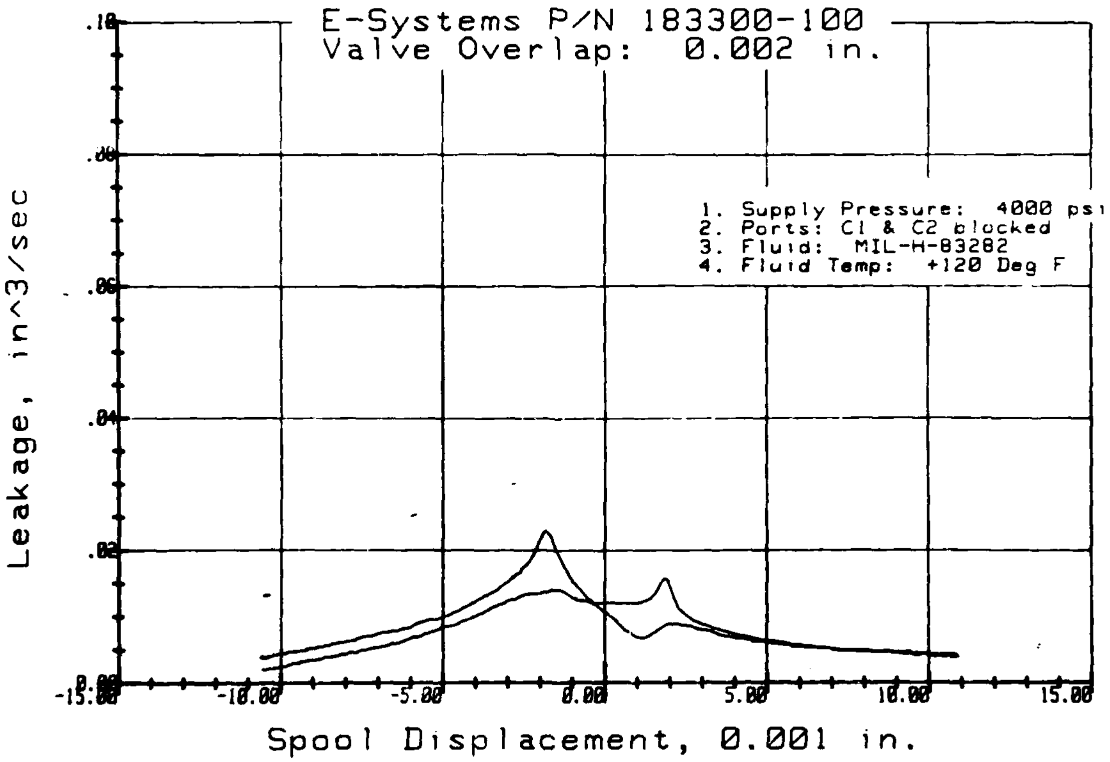
INTERNAL LEAKAGE

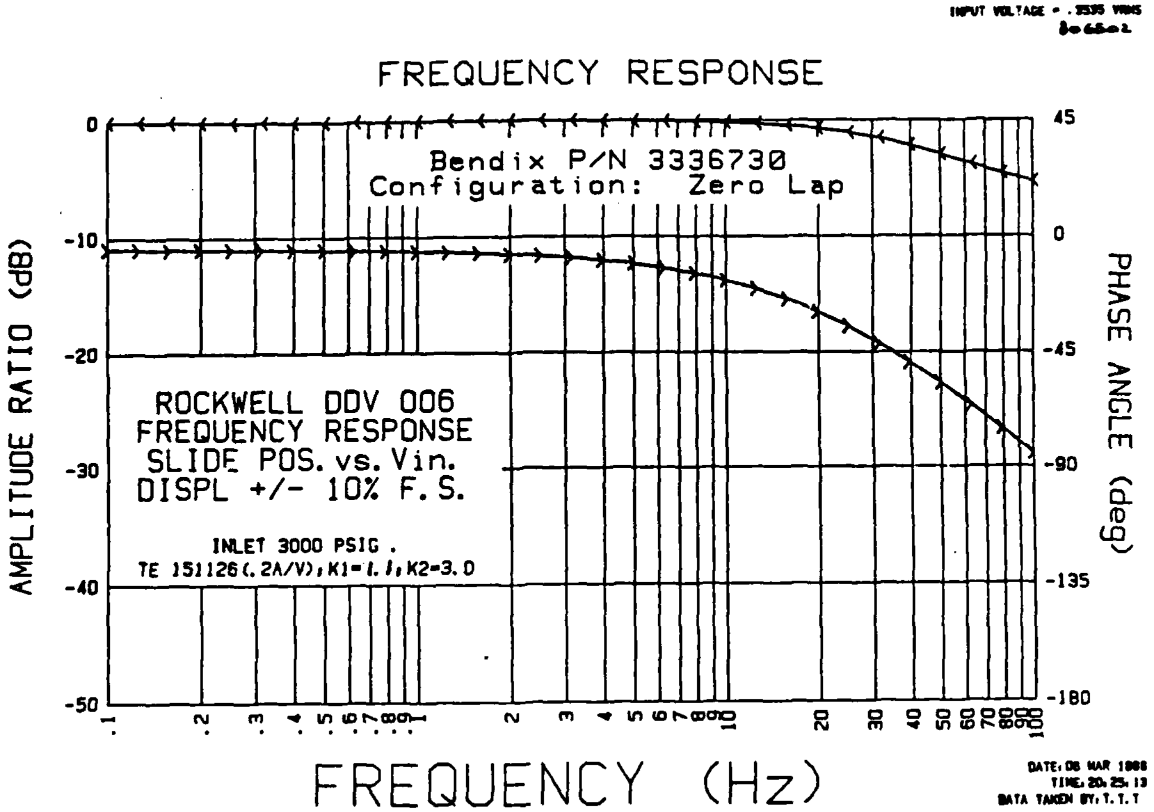


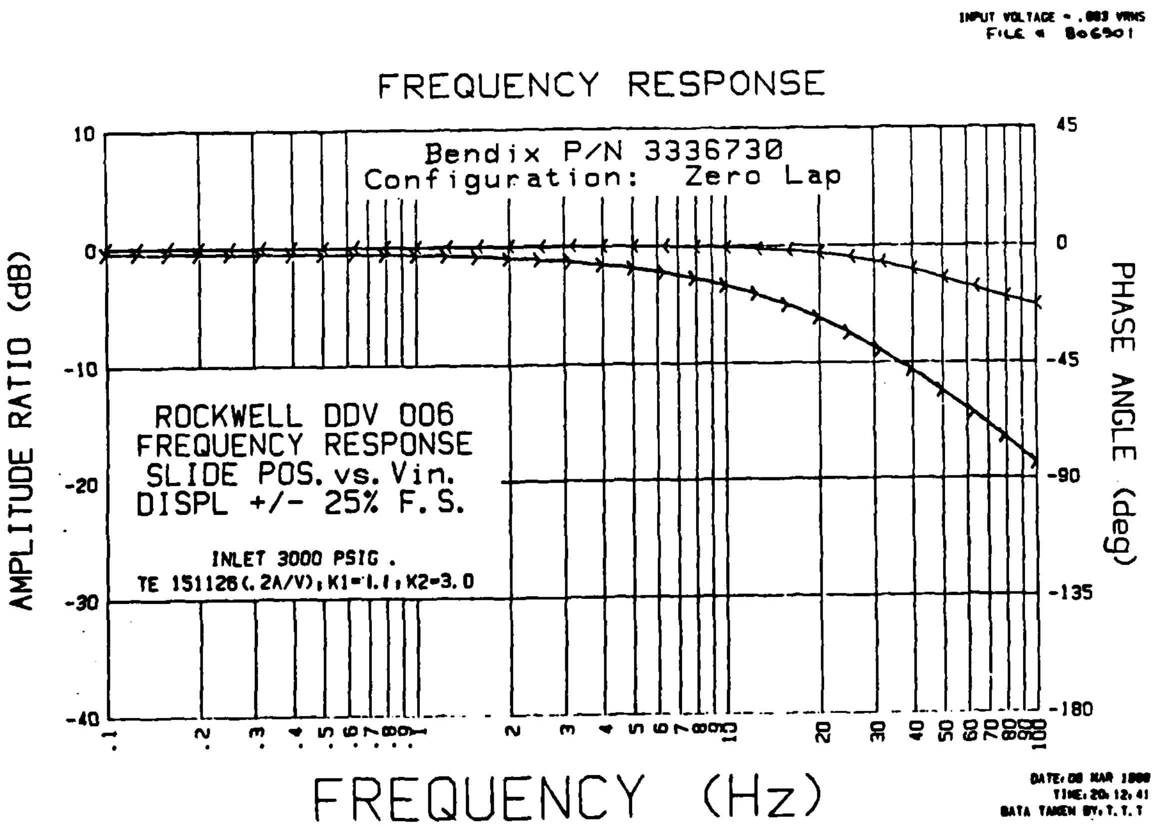


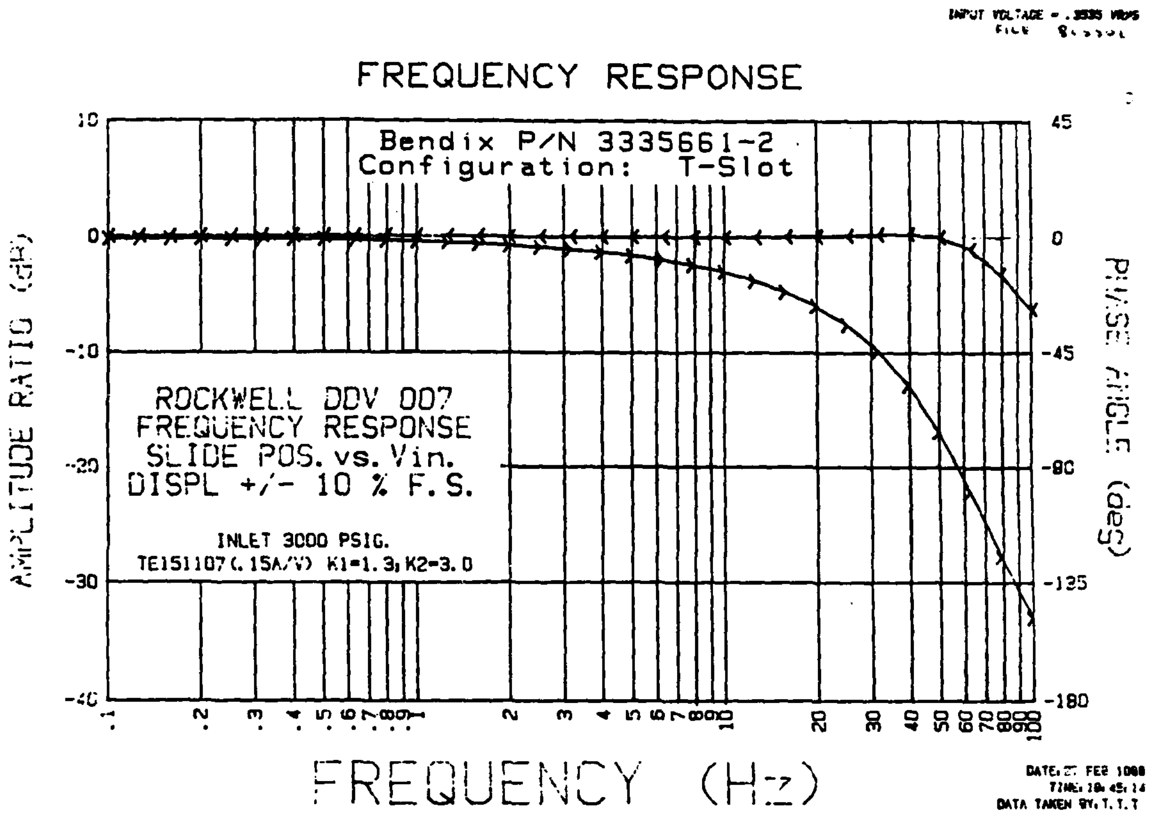
INTERNAL LEAKAGE

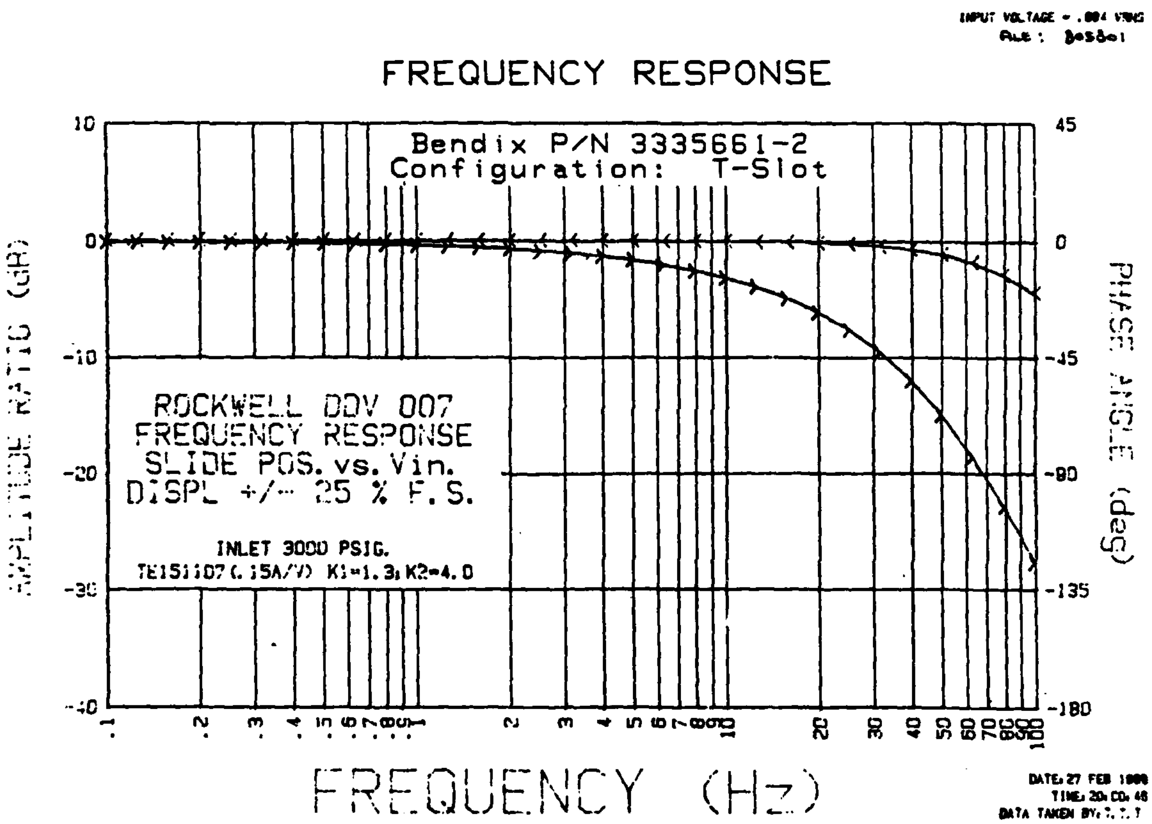


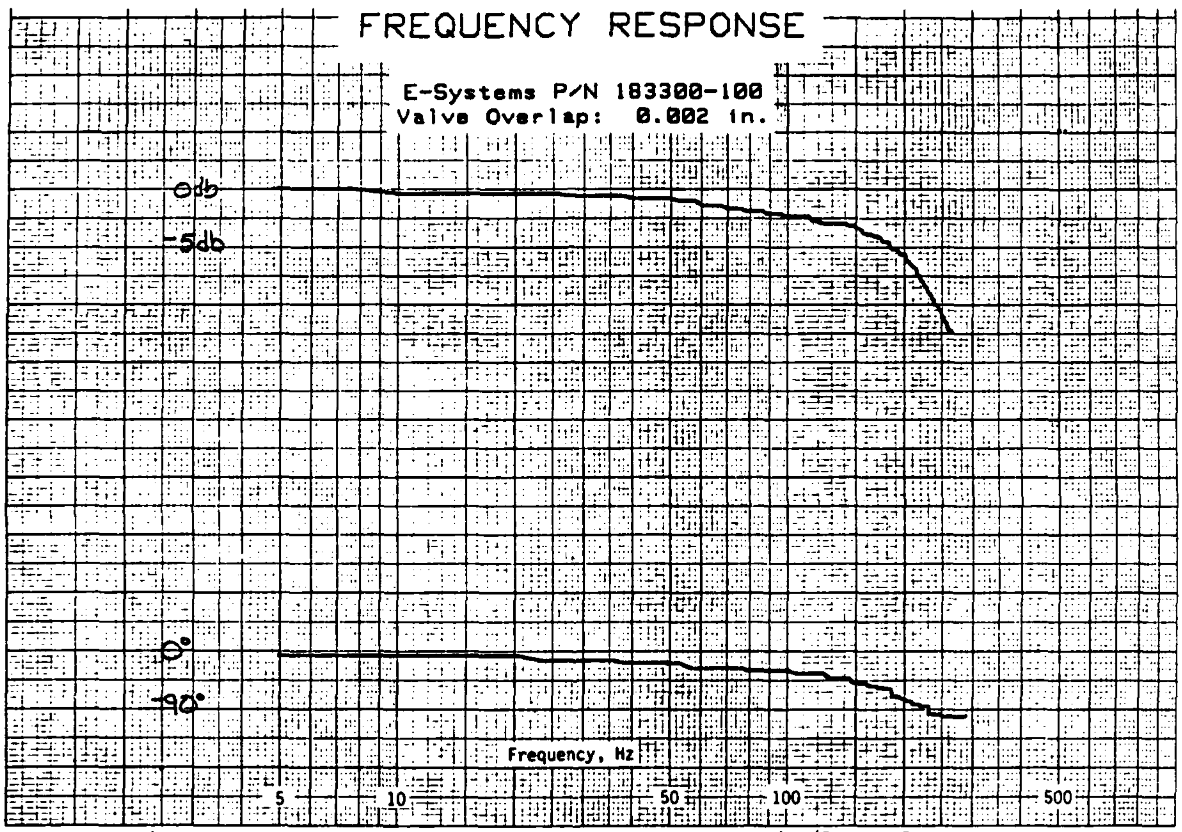



FREQUENCY RESPONSE
Bendix P/N 3336730
Configuration: Zero Lap
ROCKWELL DDV 006
FREQUENCY RESPONSE
SLIDE POS. vs. Vin.
DISPL +/- 10% F.S.
INLET 3000 PSIG .
TE 151126(.2A/V); K1=1.1; K2=3.0
AMPLITUDE RATIO (dB)
PHASE ANGLE (deg)
FREQUENCY (Hz)
DATA TAKEN BY: T.T.T


FREQUENCY RESPONSE
Bendix P/N 3336730
Configuration: Zero Lap
ROCKWELL DDV 006
FREQUENCY RESPONSE
SLIDE POS. vs. Vin.
DISPL +/- 25% F.S.
INLET 3000 PSIG .
TE 151126(.2A/V); K1=1.1; K2=3.0
AMPLITUDE RATIO (dB)
PHASE ANGLE (deg)
FREQUENCY (Hz)
DATA TAKEN BY: T.T.T


FREQUENCY RESPONSE
Bendix P/N 3335661-2
Configuration: T-Slot
ROCKWELL DDV 007
FREQUENCY RESPONSE
SLIDE POS. vs. Vin.
DISPL +/- 10 % F.S.
INLET 3000 PSIG.
TE151107(.15A/V) K1=1.3; K2=3.0
AMPLITUDE RATIO (dB)
PHASE ANGLE (deg)
FREQUENCY (Hz)
10
0
-10
-20
-30
-40
45
0
-45
-90
-135
-180


FREQUENCY RESPONSE
Bendix P/N 3335661-2
Configuration: T-Slot
ROCKWELL DDV 007
FREQUENCY RESPONSE
SLIDE POS. vs. Vin.
DISPL +/- 25 % F.S.
INLET 3000 PSIG.
TE151107(.15A/V) K1=1.3; K2=4.0
AMPLITUDE RATIO (dB)
PHASE ANGLE (deg)
FREQUENCY (Hz)
10
0
-10
-20
-30
-40
45
0
-45
-90
-135
-180



± 25% STROKE OVERLAP SET UP 8000psi CTFE 11/30/87 ROCKWELL VALVE

APPENDIX C

SERVO ACTUATOR TEST DATA

Contents | Page

Dynamic Response Tests

Step response, E-Systems zero lap valve, no load 128
Step response, E-Systems zero lap valve, loaded 129
Step response, E-Systems overlapped valve, no load 130
Step response, E-Systems overlapped valve, loaded 131
Frequency response, E-Systems zero lap valve, no load 132
Frequency response, E-Systems zero lap valve, loaded 133
Frequency response, E-Systems overlapped valve, no load 134
Frequency response, E-Systems overlapped valve, loaded 135

Pressure Level Switching Tests

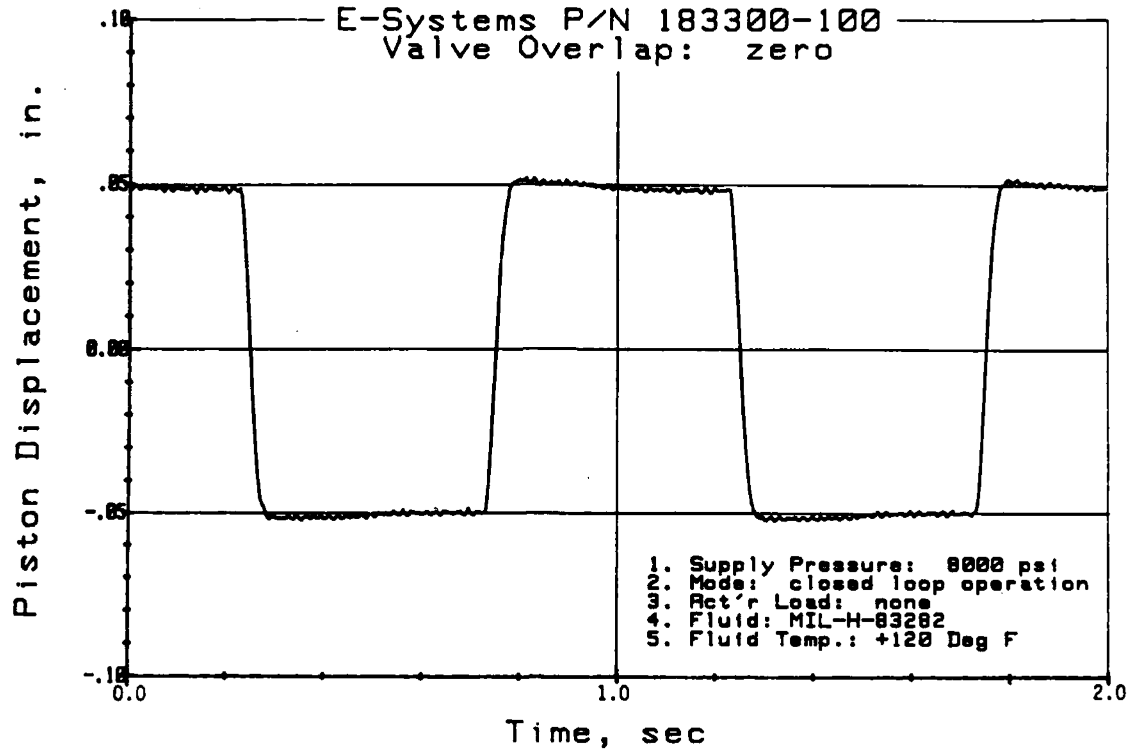
Actuator Piston Motionless
E-Systems zero lap valve, 3400 rpm pump speed 136
E-Systems zero lap valve, 5900 rpm pump speed 136
Actuator Piston Moving
E-Systems zero lap valve, 0^{o} switching point 137
E-Systems zero lap valve, 90^{o} switching point 138
E-Systems overlapped valve, 0^{o} switching point 139
E-Systems overlapped valve, 90^{o} switching point 140

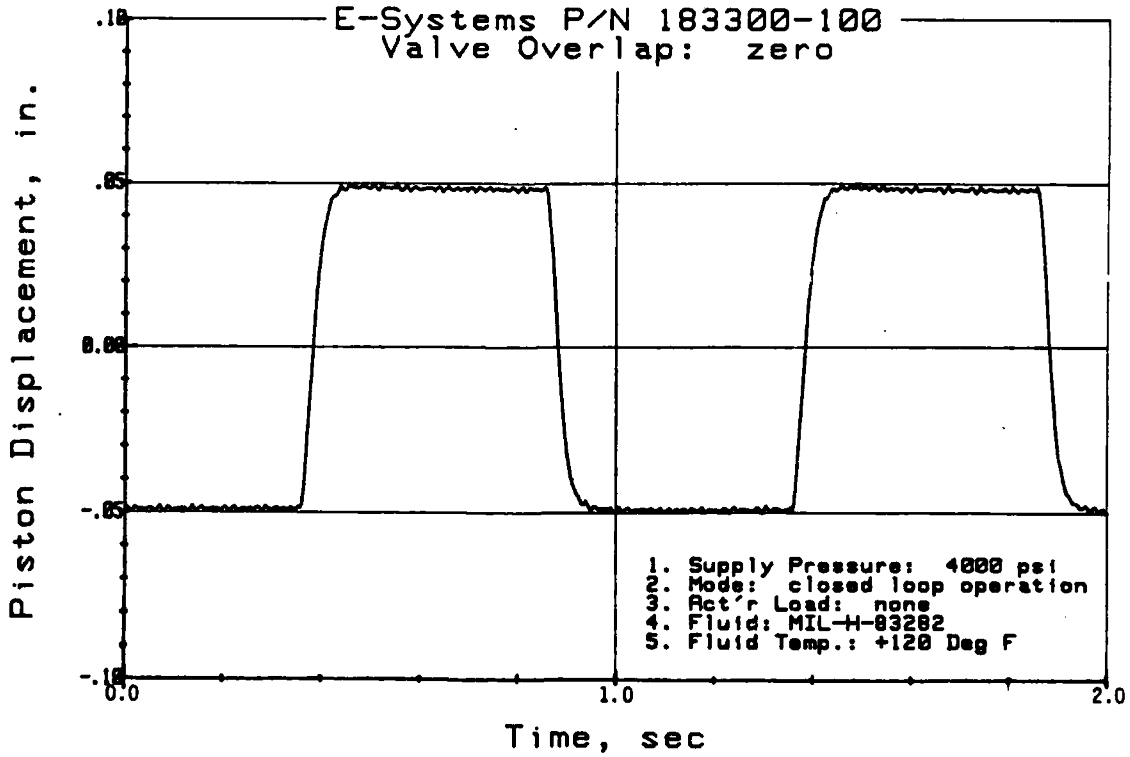
Energy Consumption Tests

E-Systems zero lap valve
Pump, actuator not loaded 141
Actuator, actuator not loaded 142
System, actuator not loaded 143
Pump, actuator loaded 144
Actuator, actuator loaded 145
System, actuator loaded 146

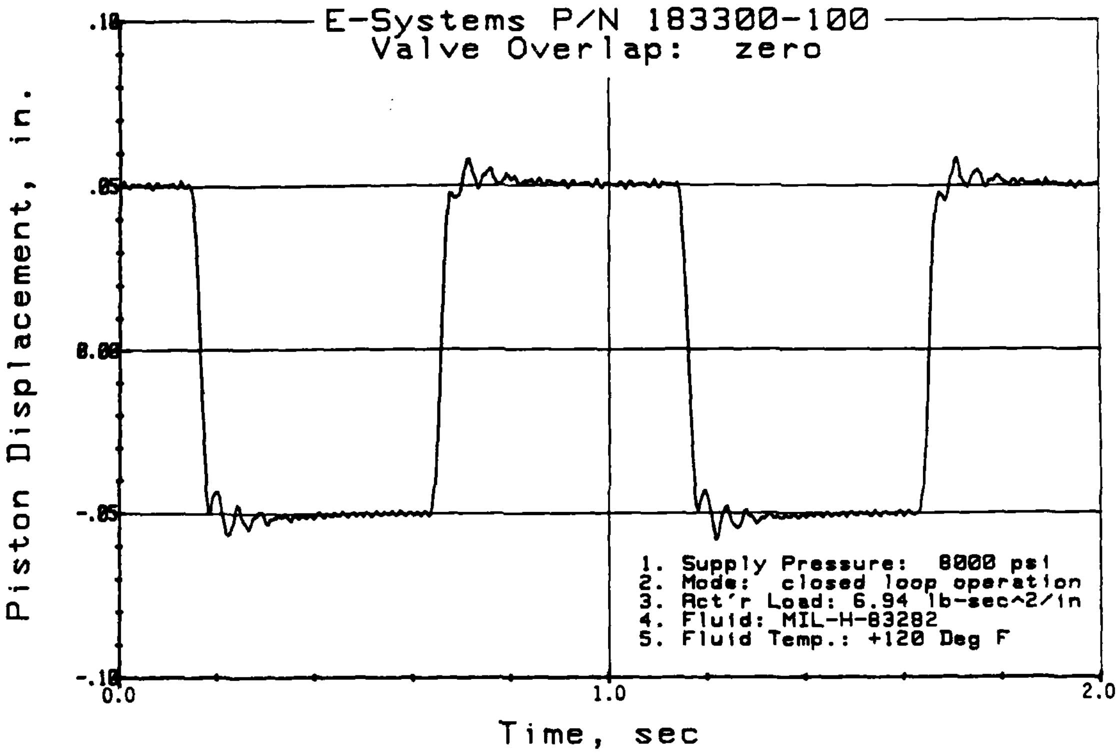
E-Systems overlapped valve
Pump, actuator not loaded 147
Actuator, actuator not loaded 148
System, actuator not loaded 149
Pump, actuator loaded 150
Actuator, actuator loaded 151
System, actuator loaded 152

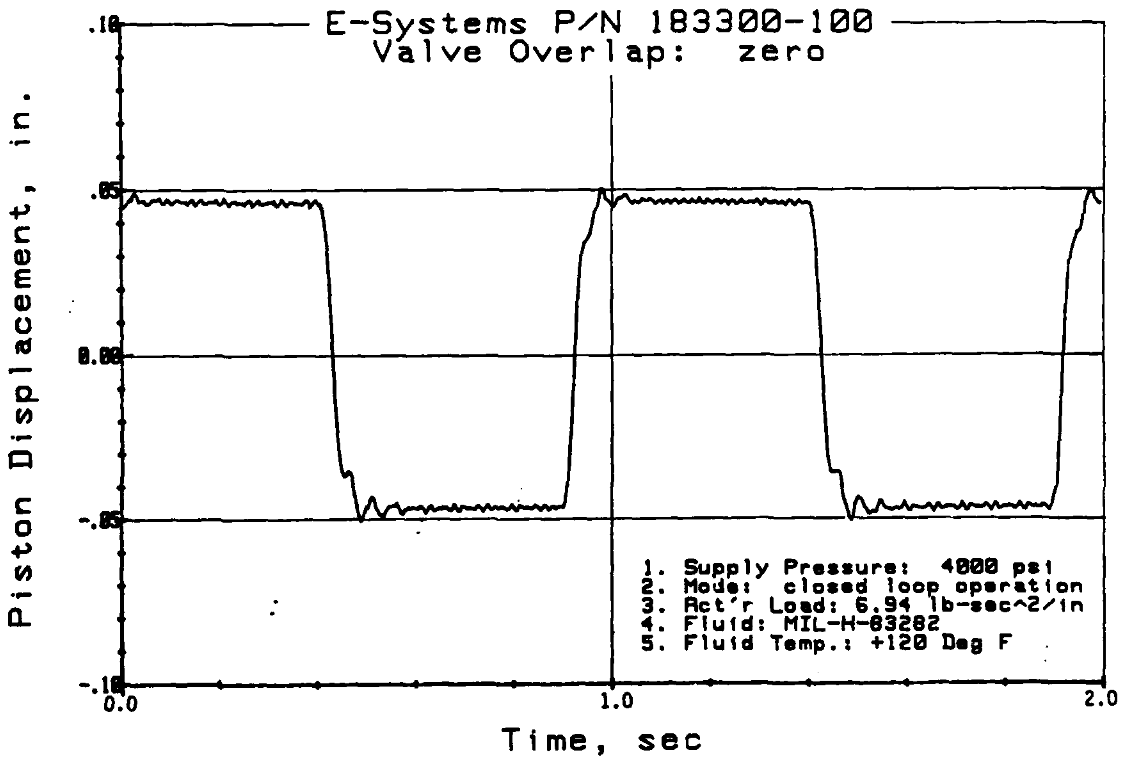
STEP RESPONSE



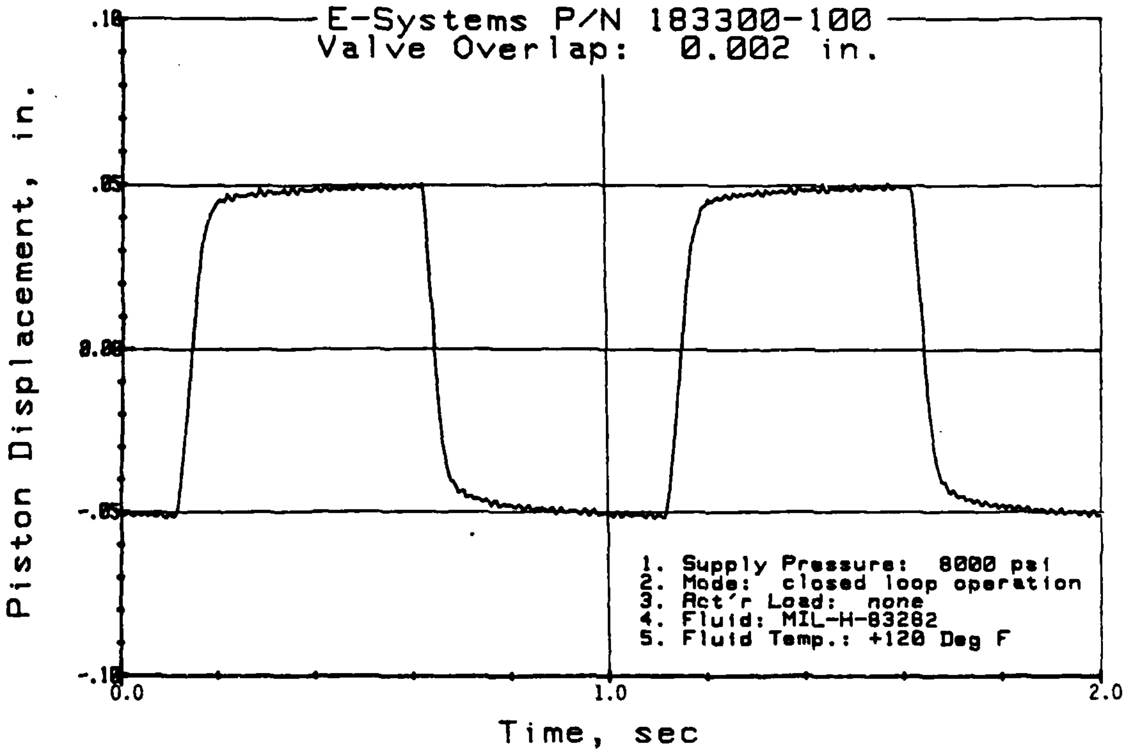


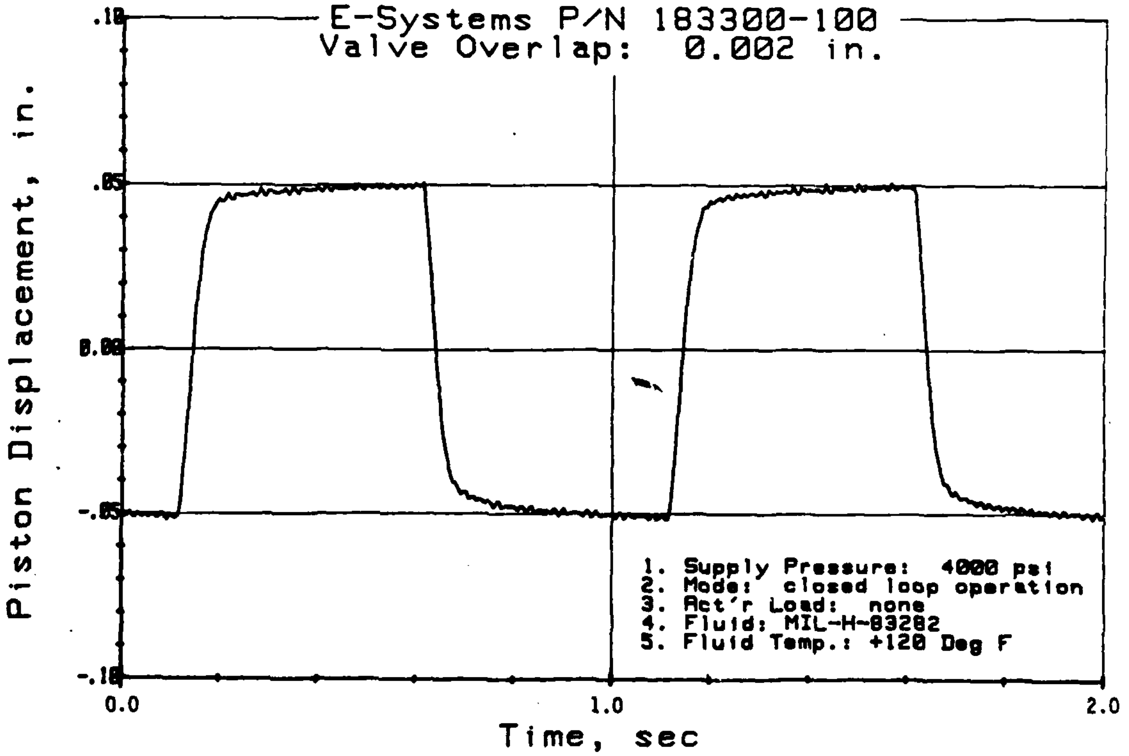
STEP RESPONSE



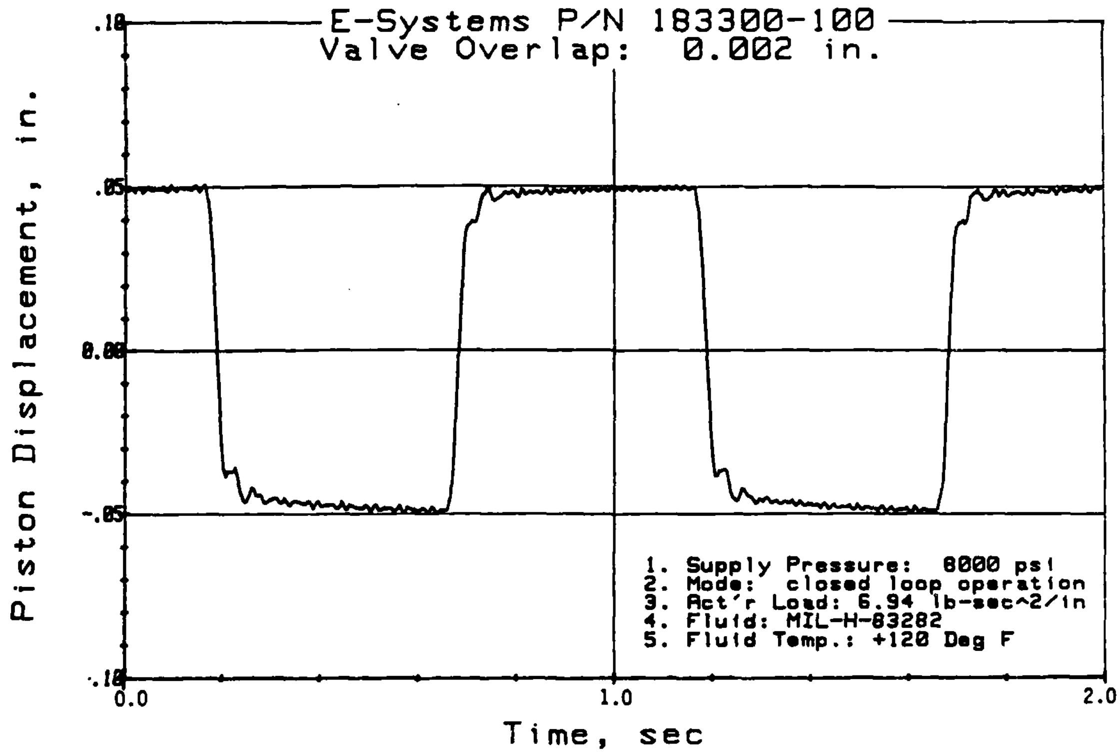


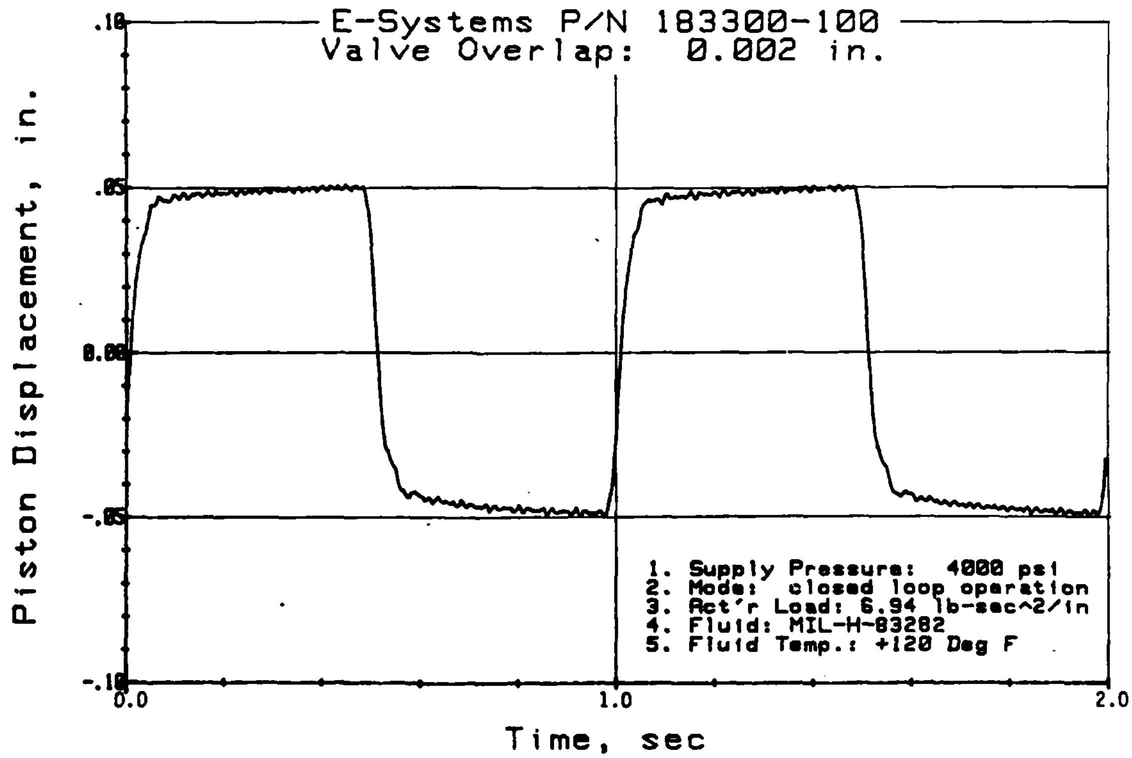
STEP RESPONSE



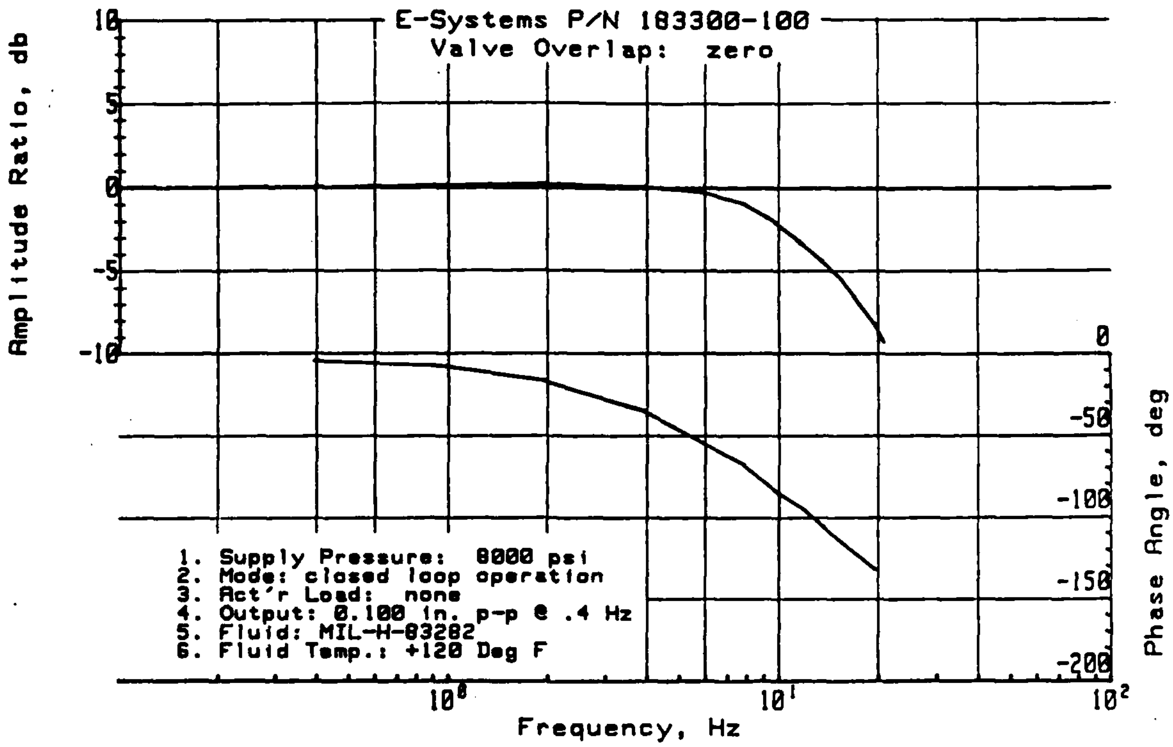


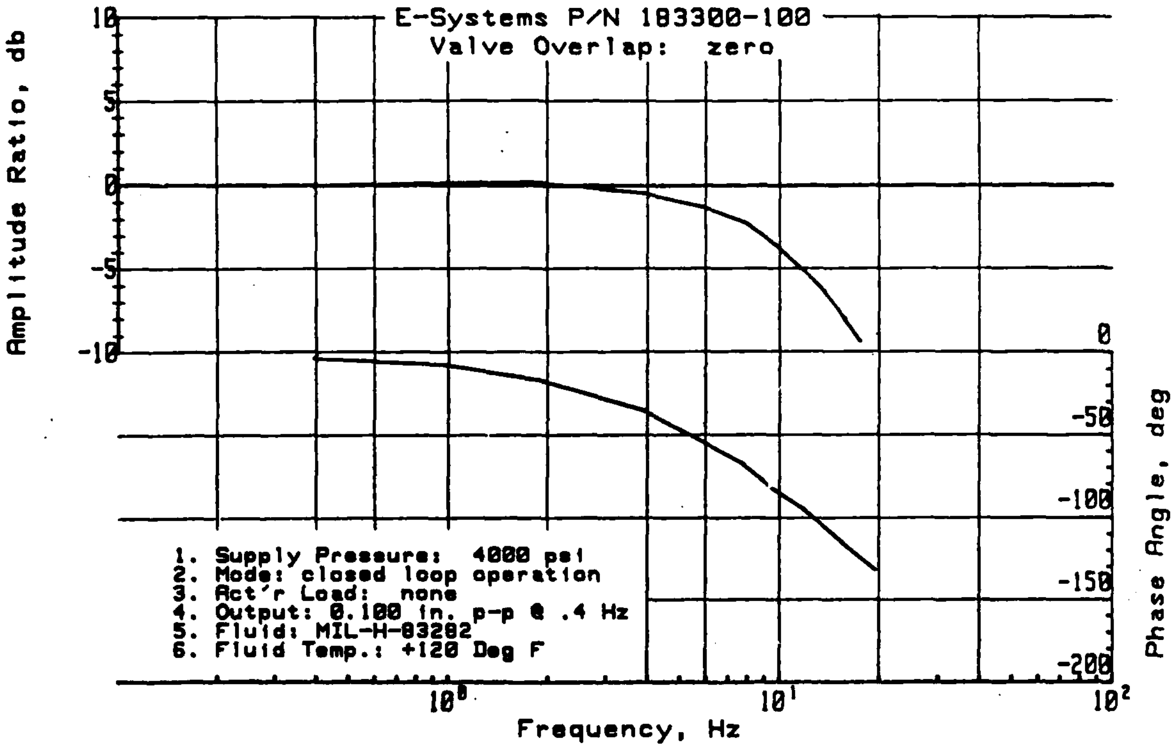
STEP RESPONSE



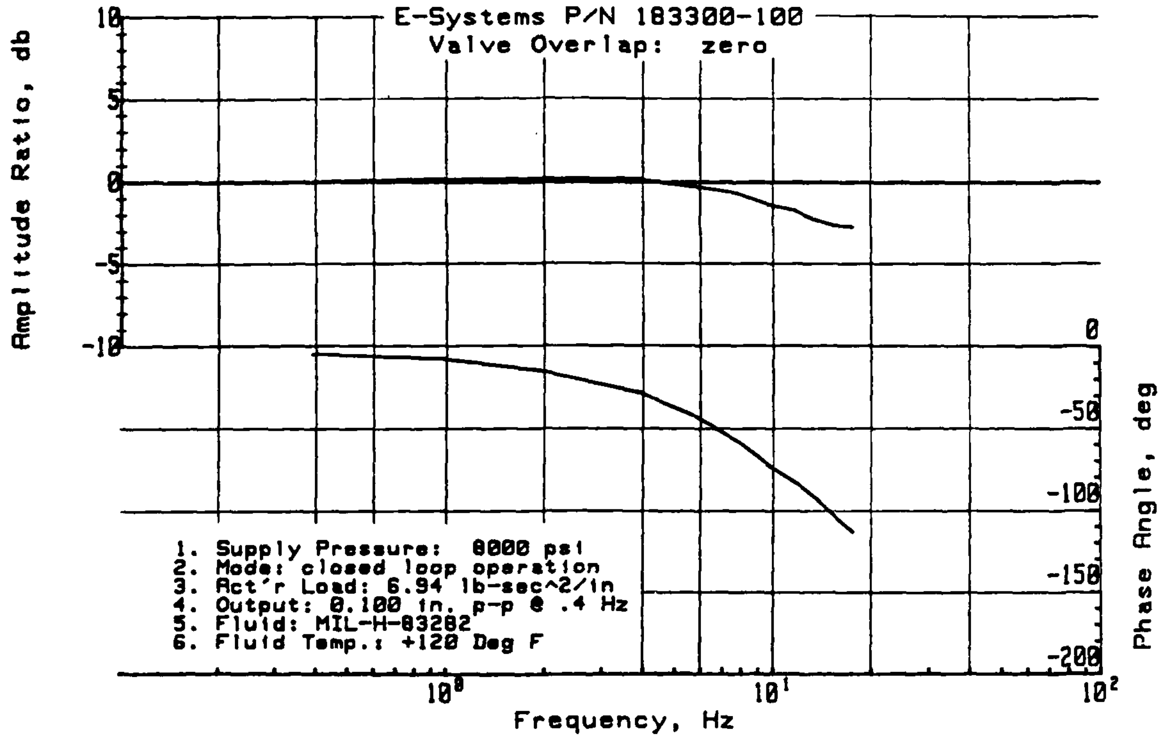


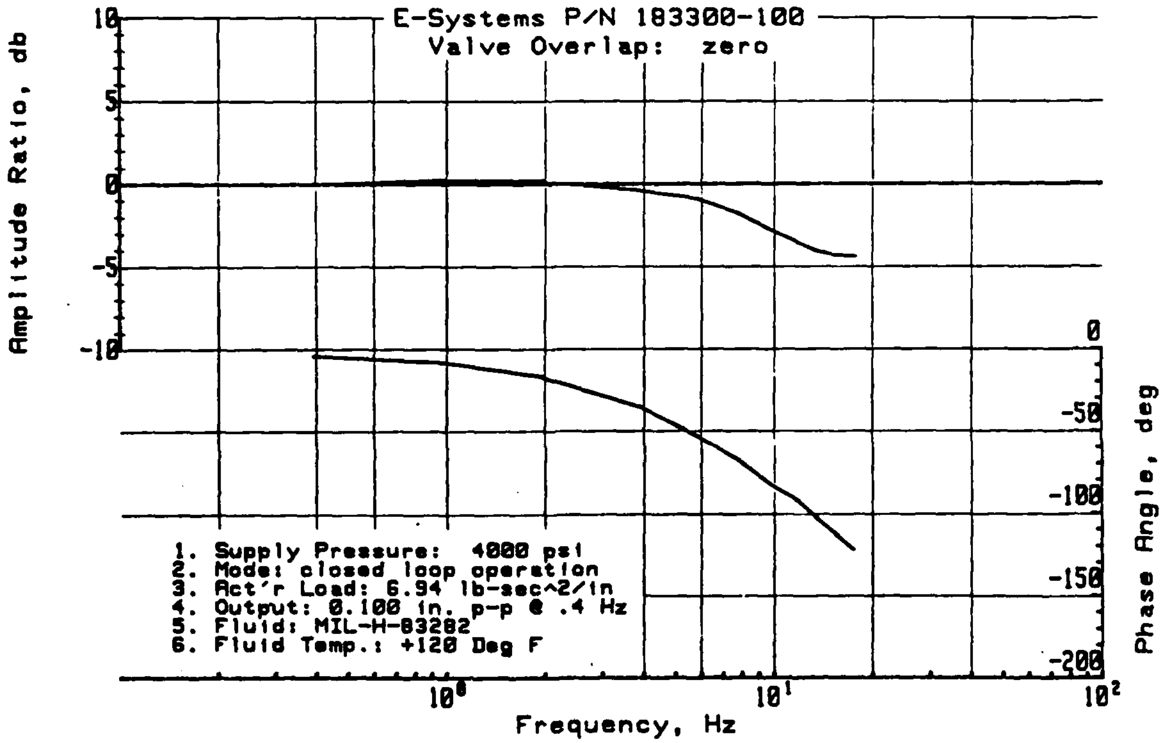
FREQUENCY RESPONSE



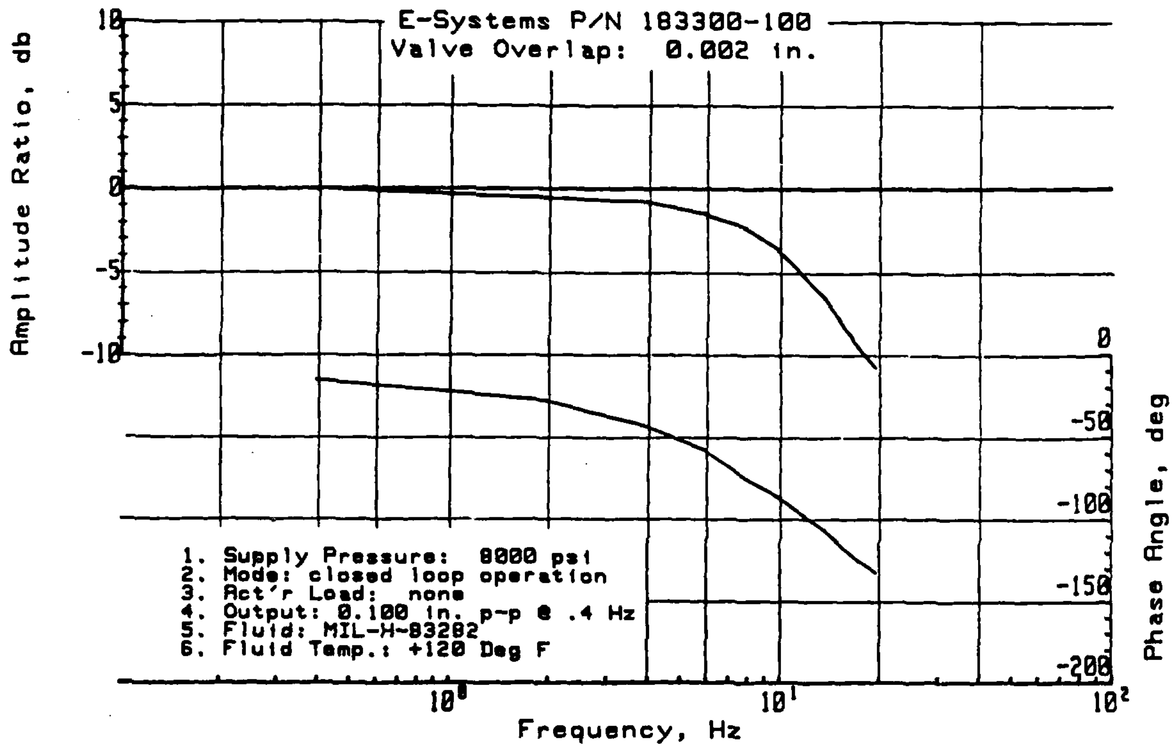


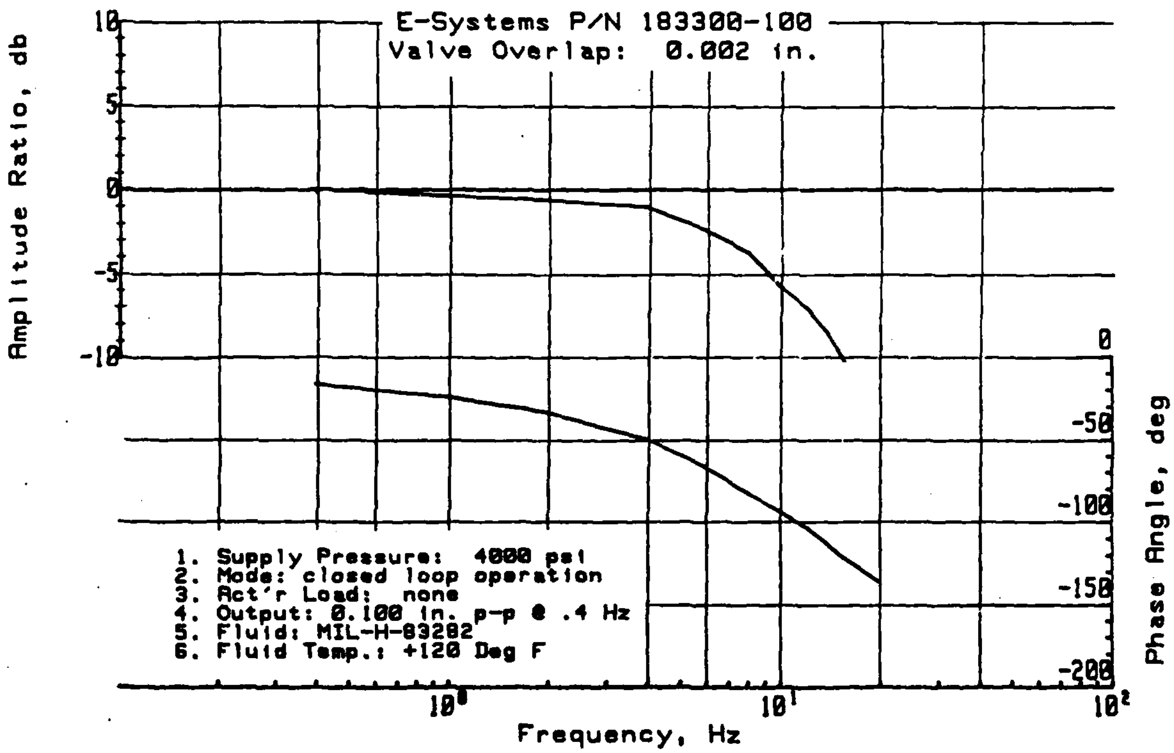
FREQUENCY RESPONSE



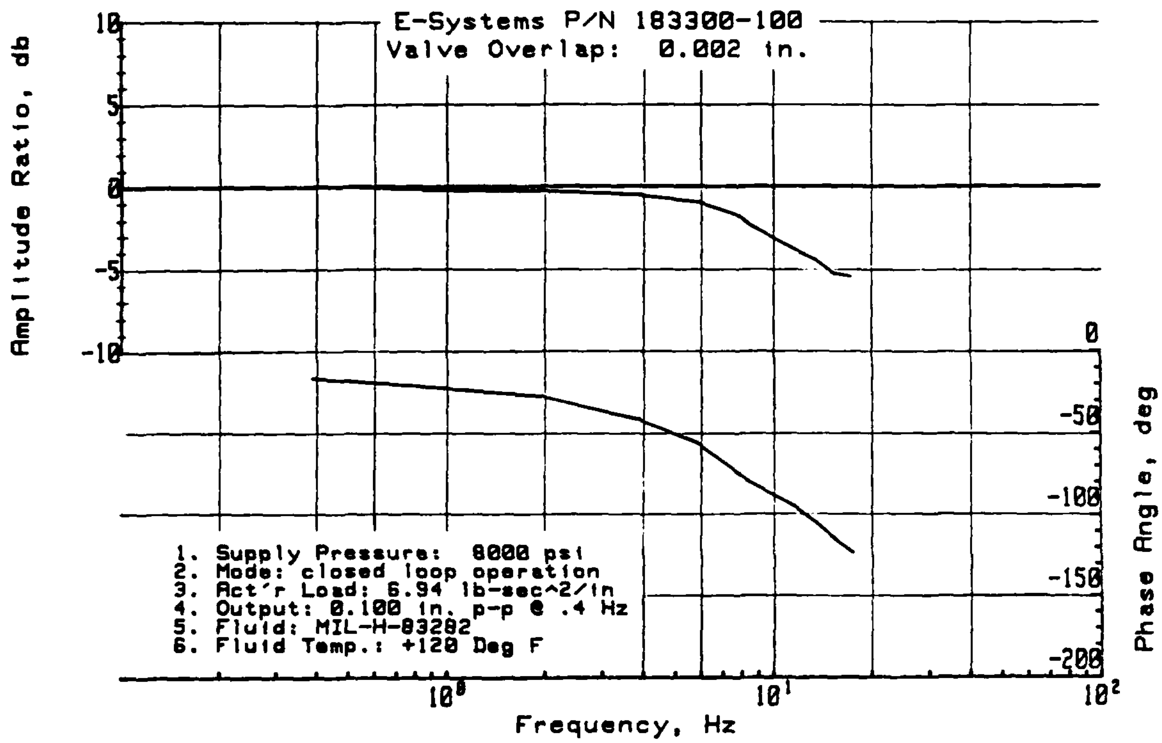


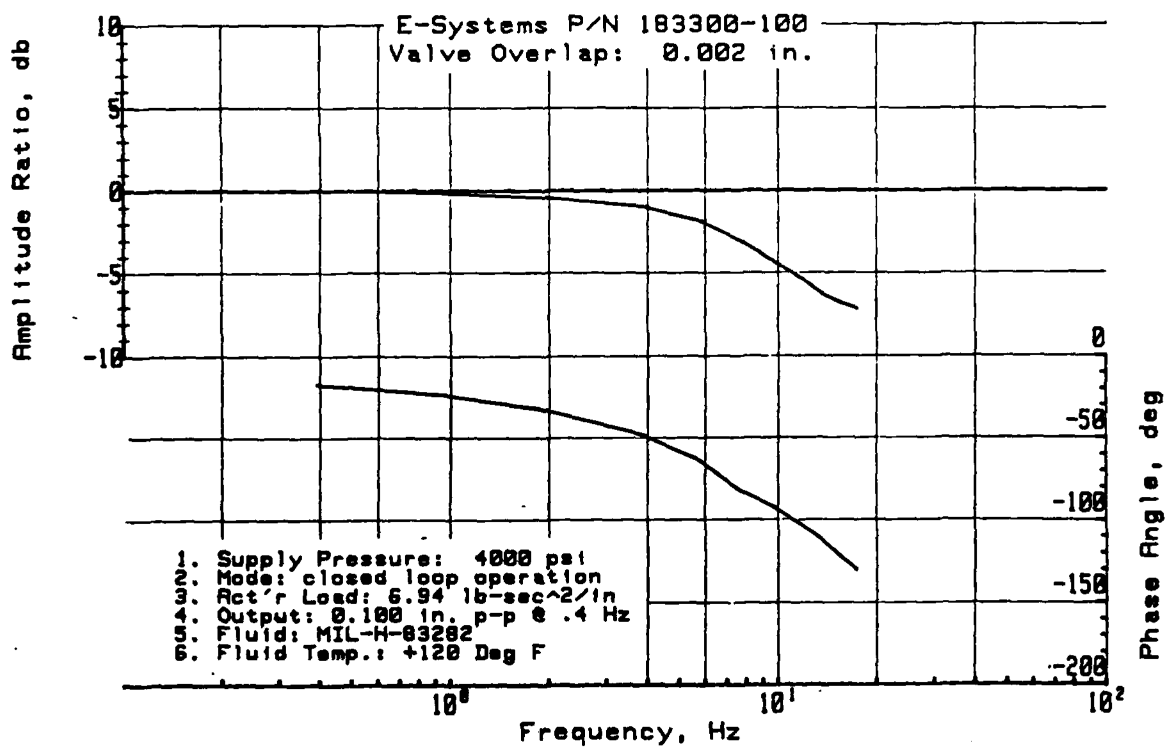
FREQUENCY RESPONSE



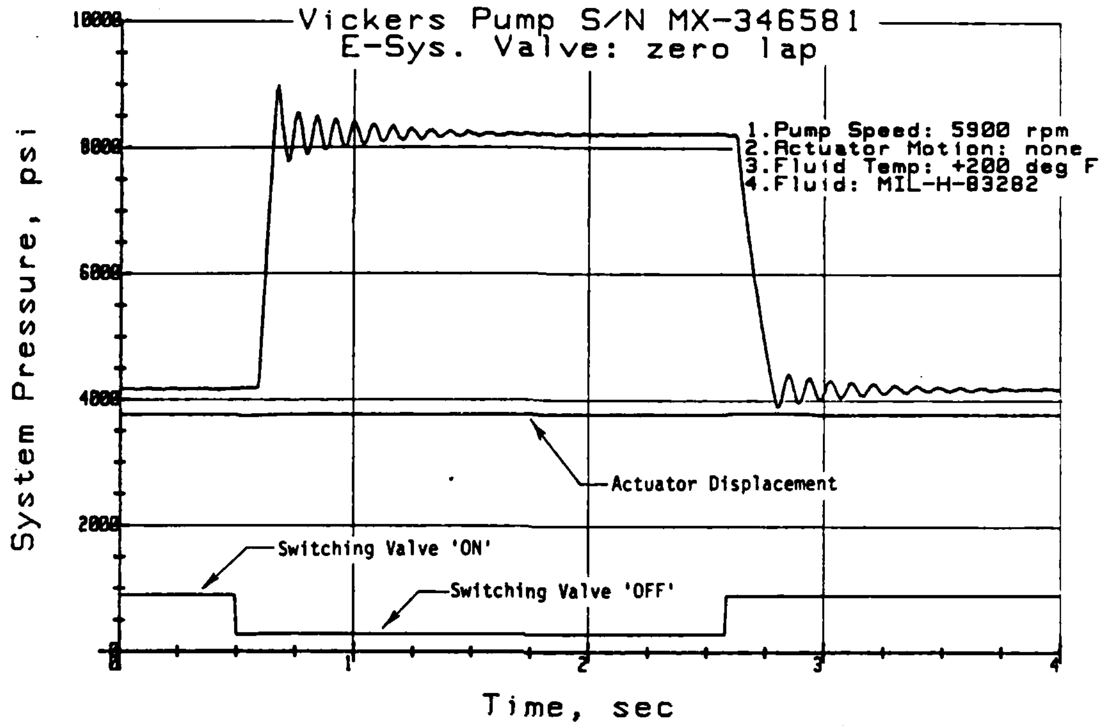


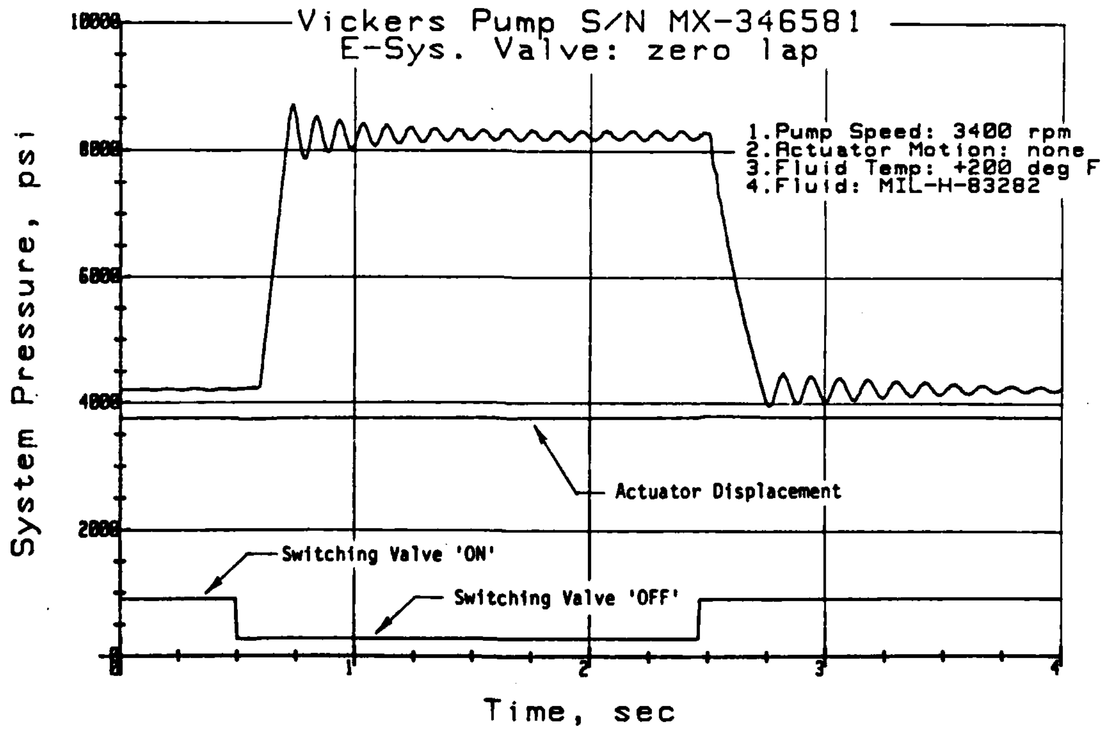
FREQUENCY RESPONSE



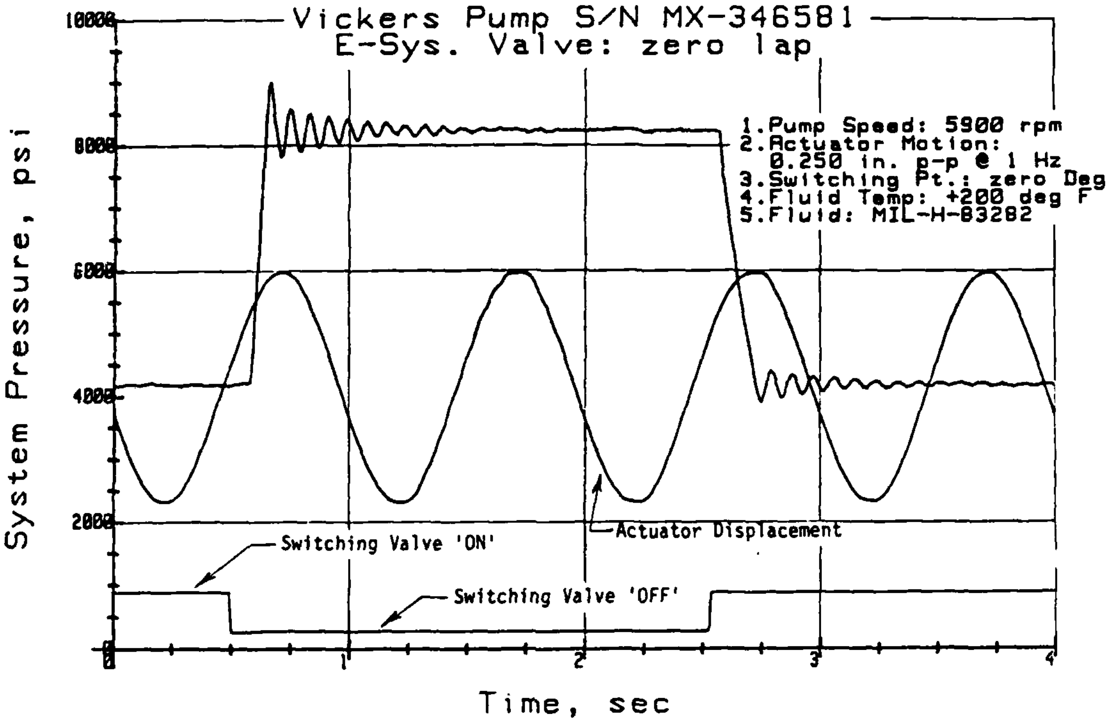


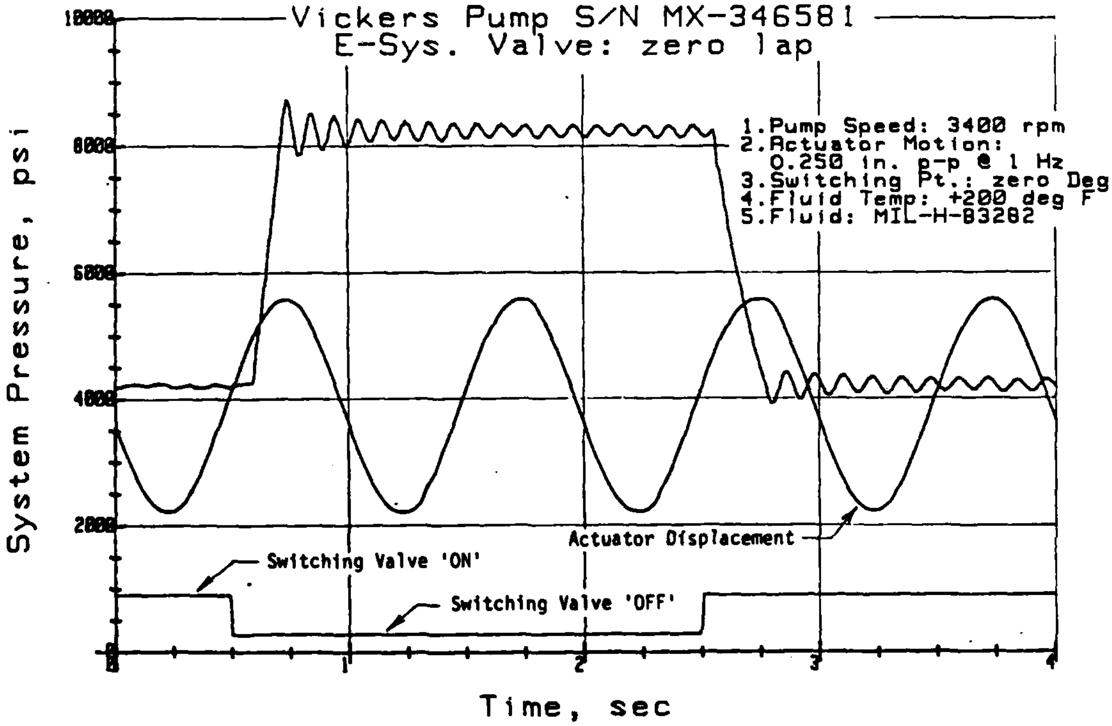
PRESSURE LEVEL SWITCHING



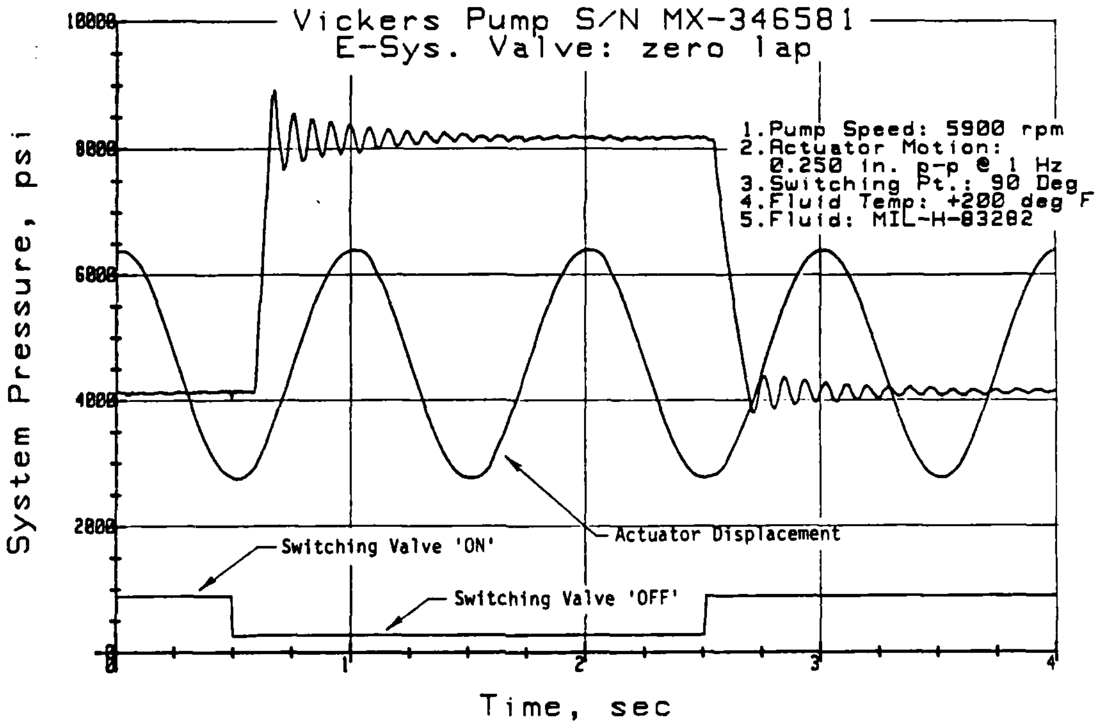


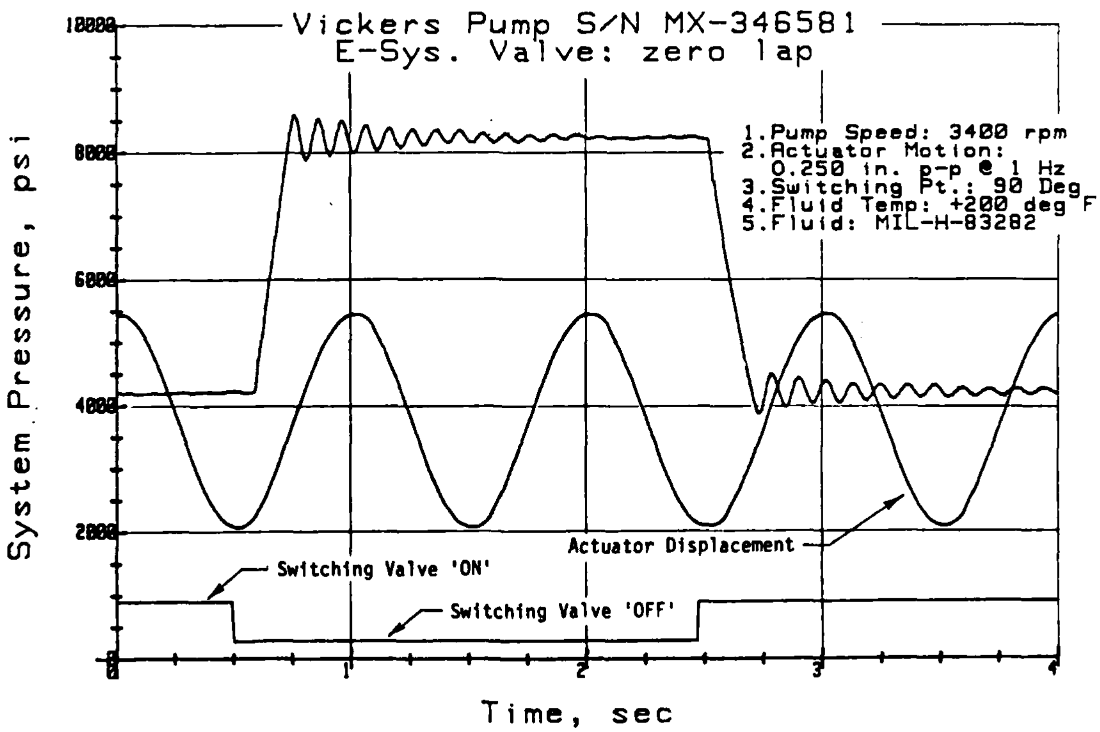
PRESSURE LEVEL SWITCHING



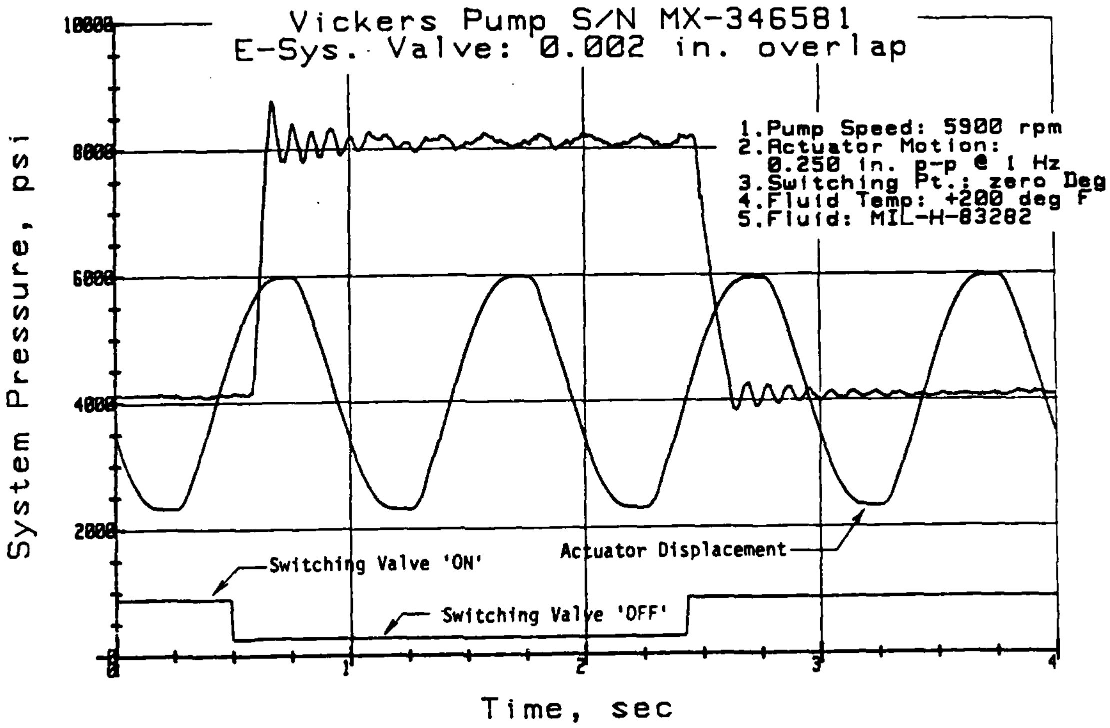


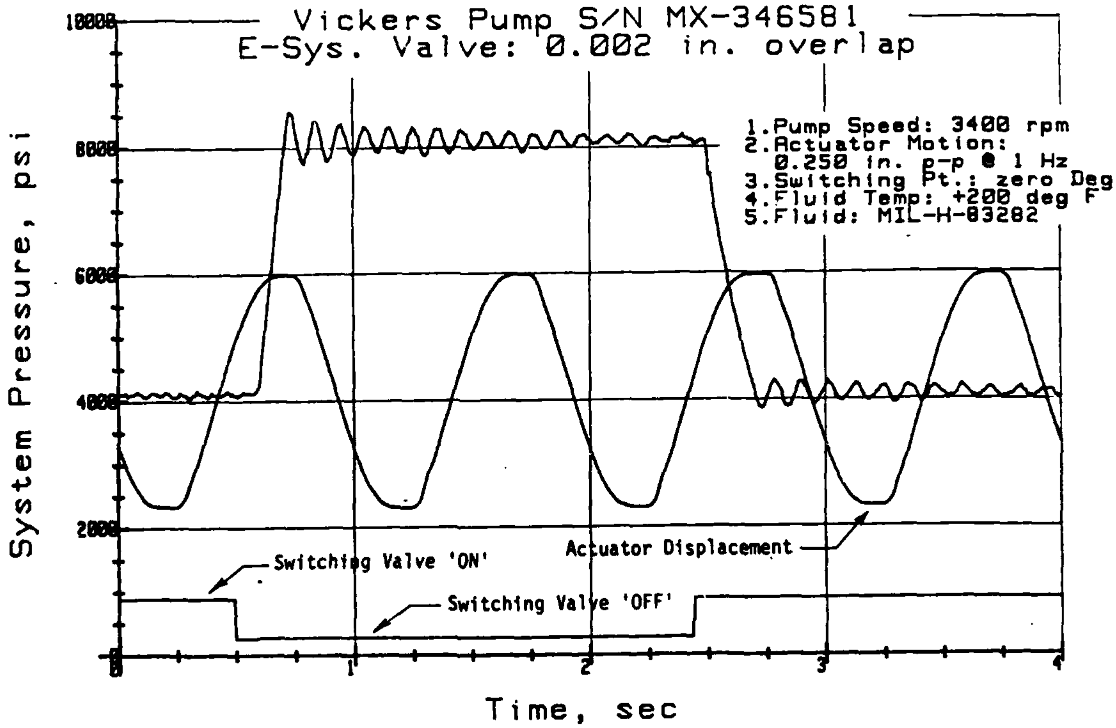
PRESSURE LEVEL SWITCHING



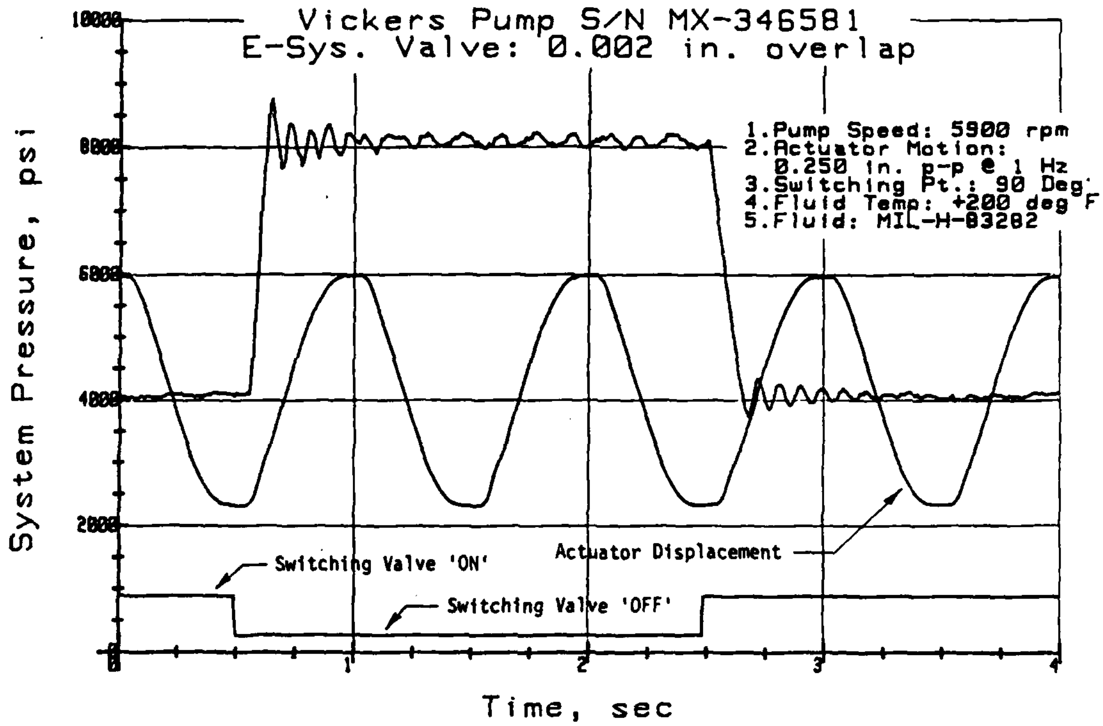


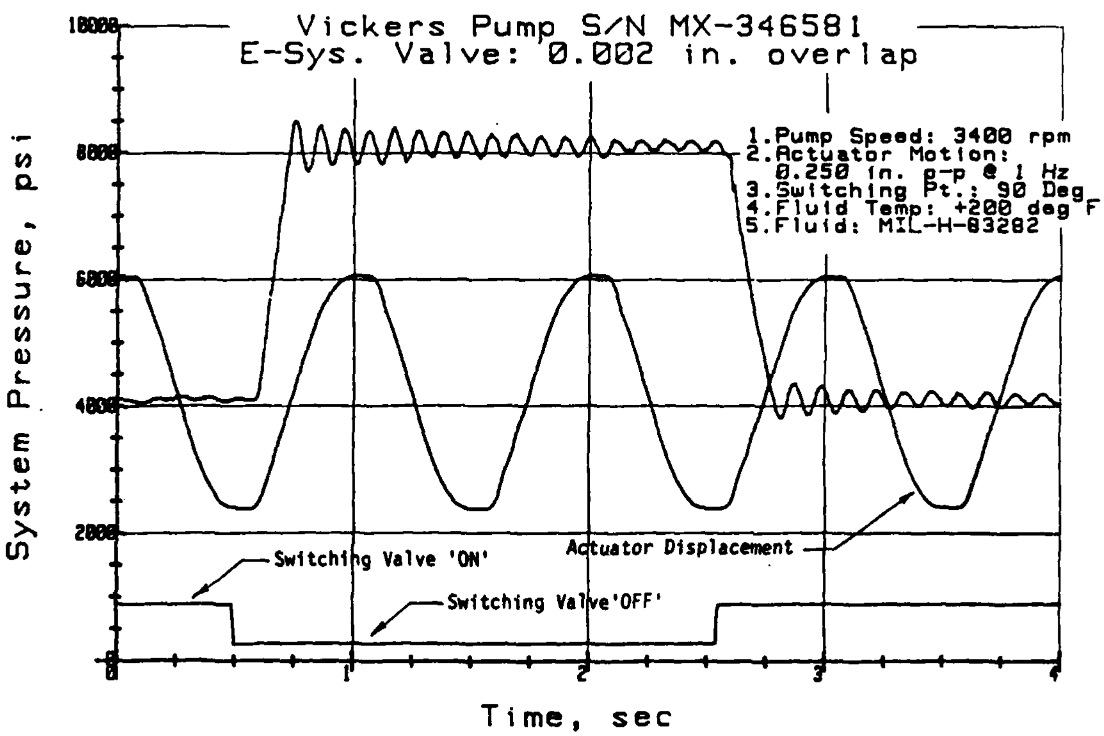
PRESSURE LEVEL SWITCHING





PRESSURE LEVEL SWITCHING





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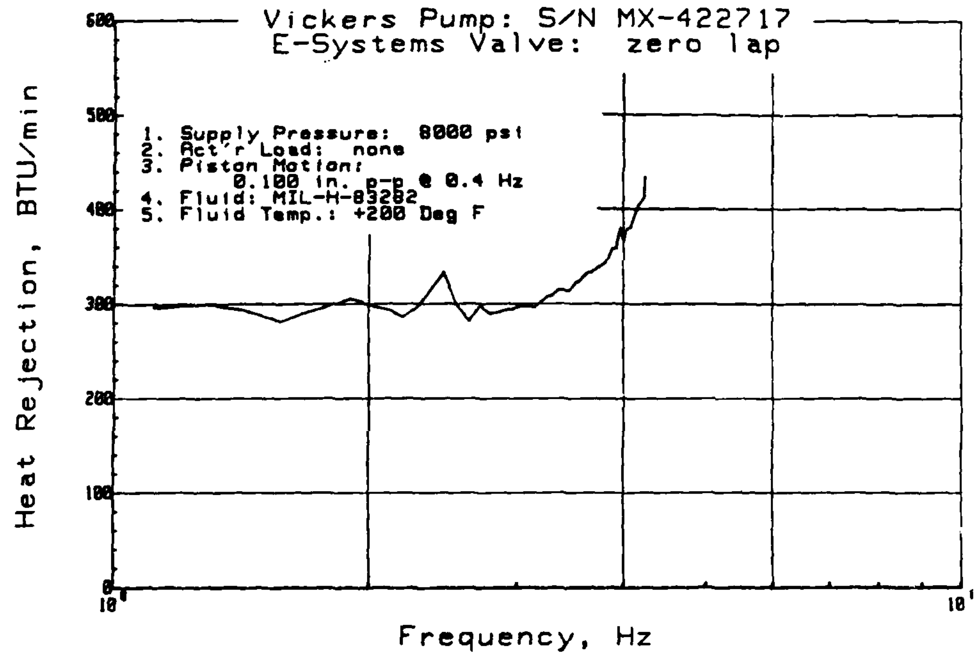

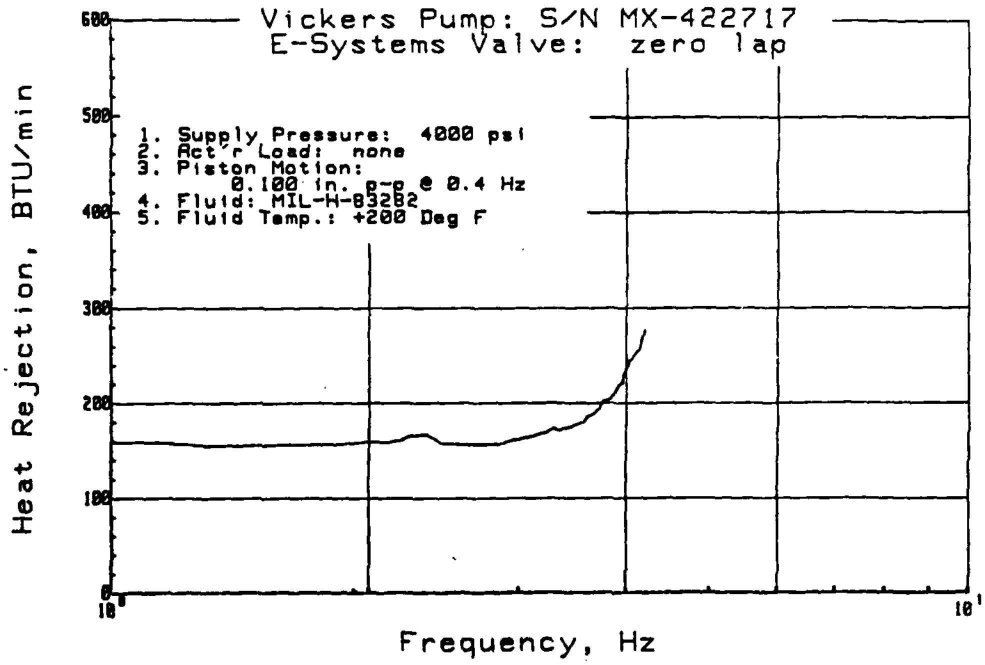

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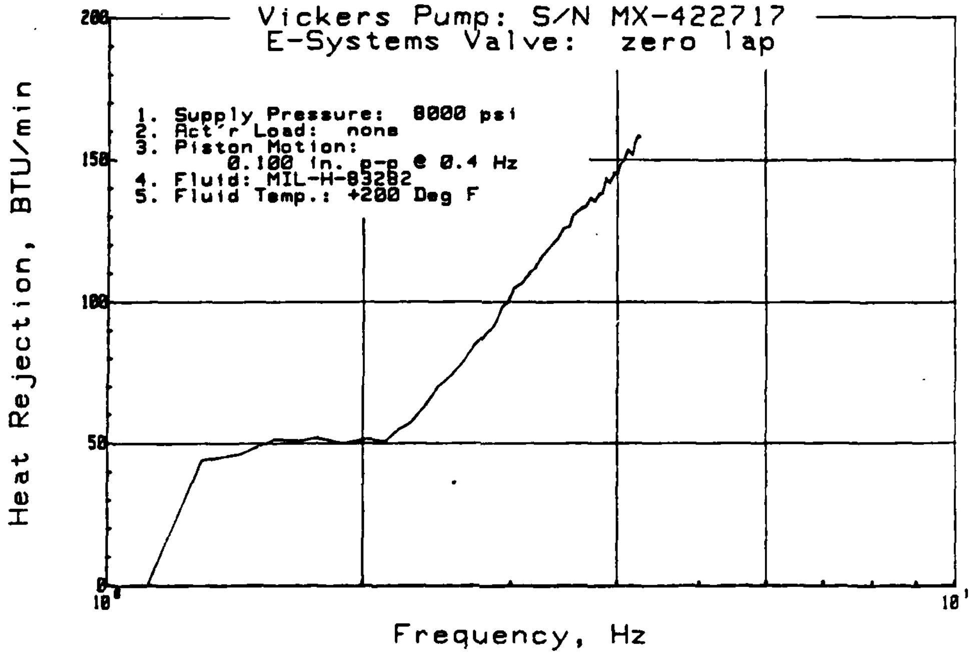

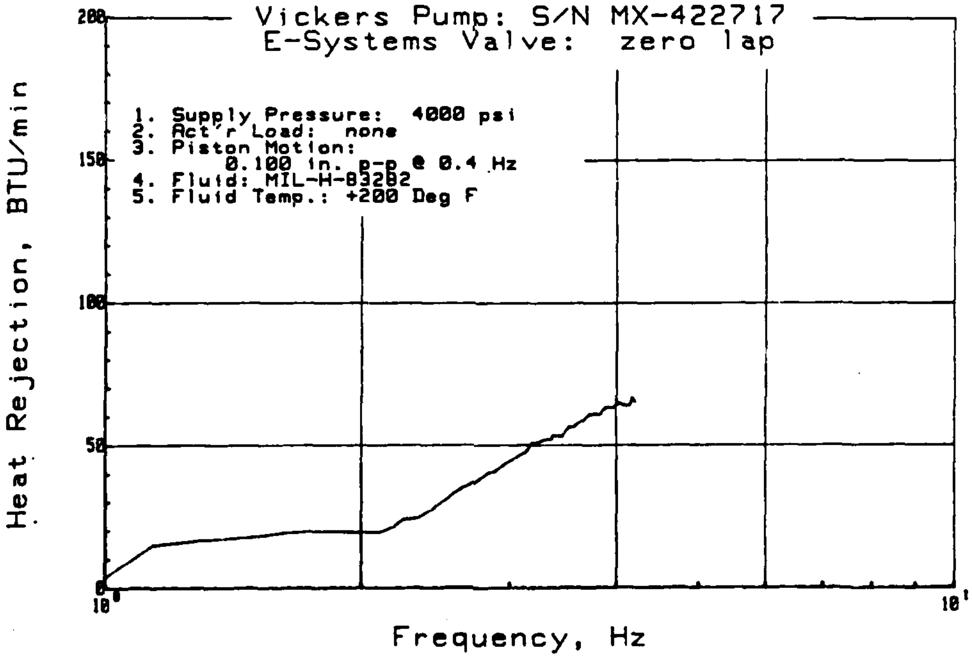

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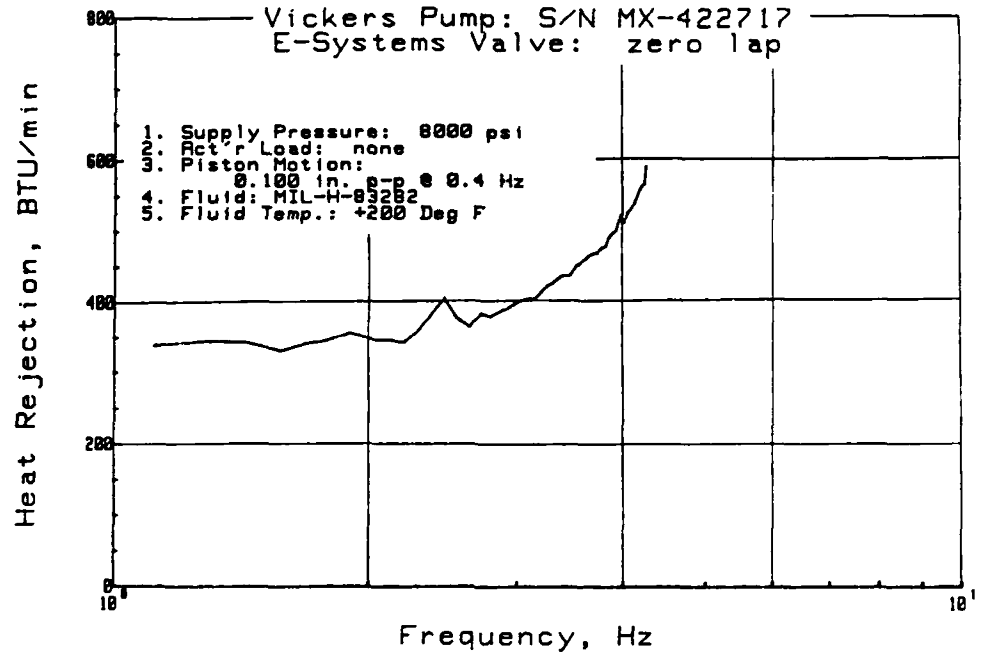

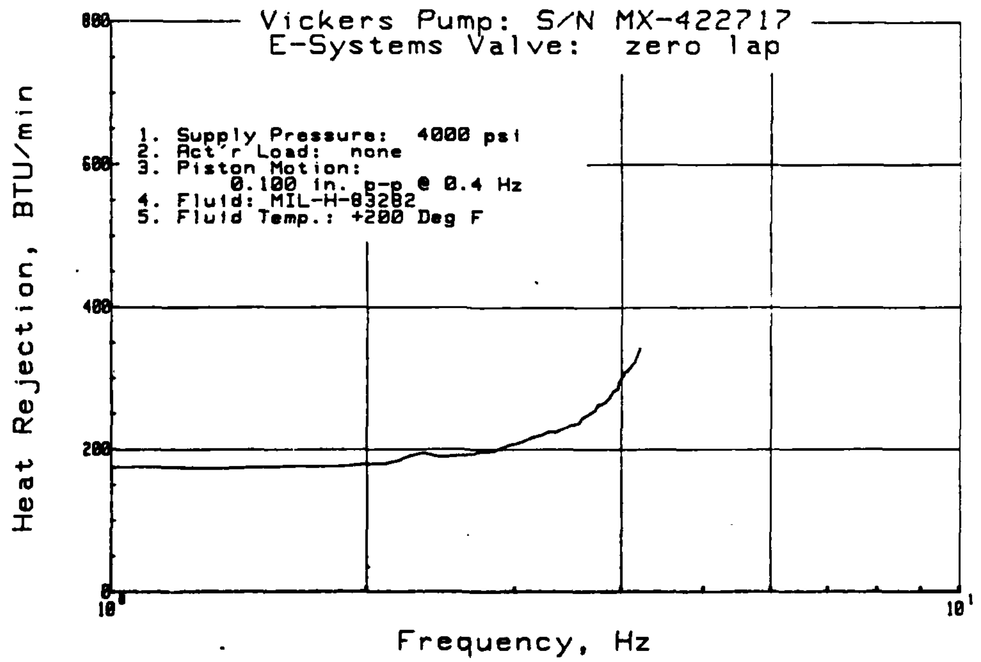

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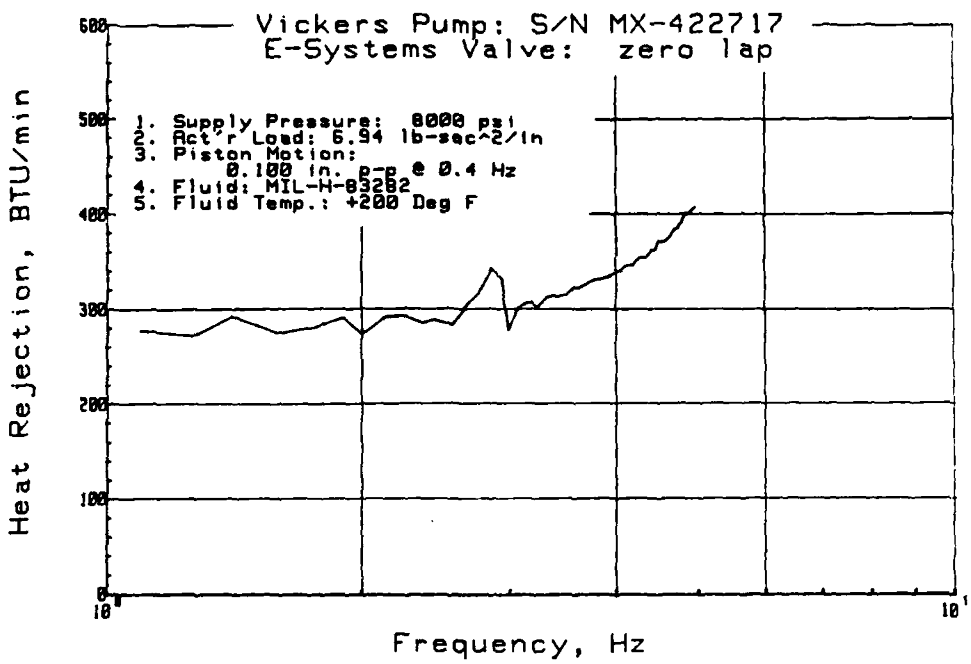

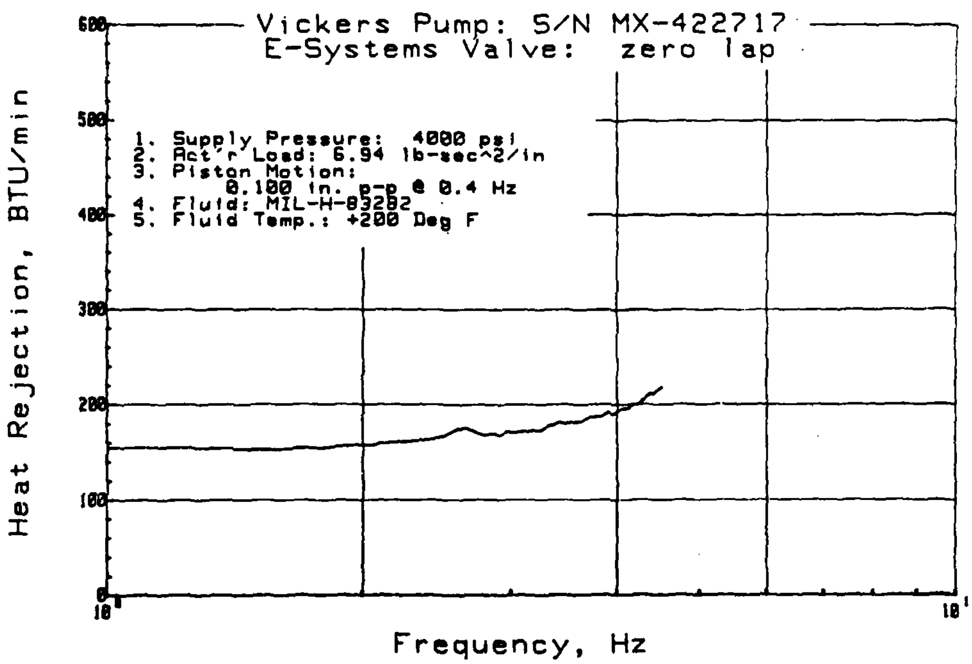

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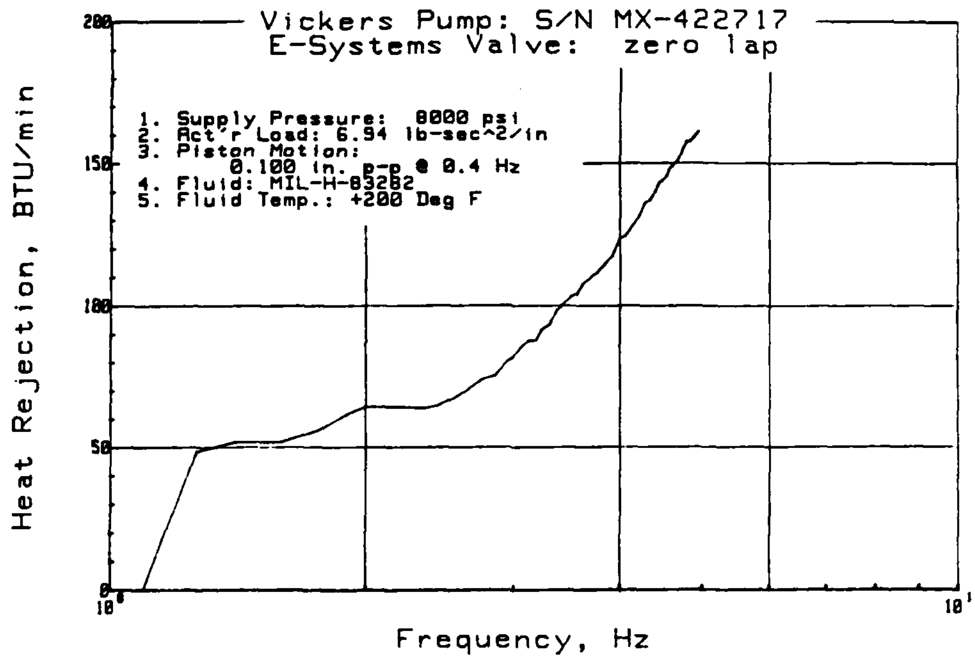

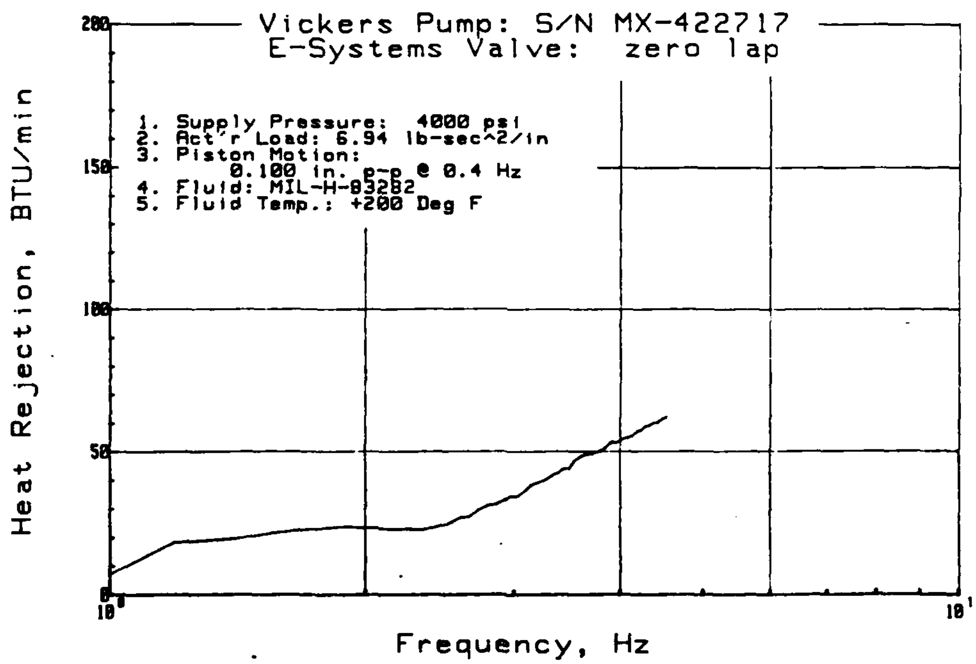

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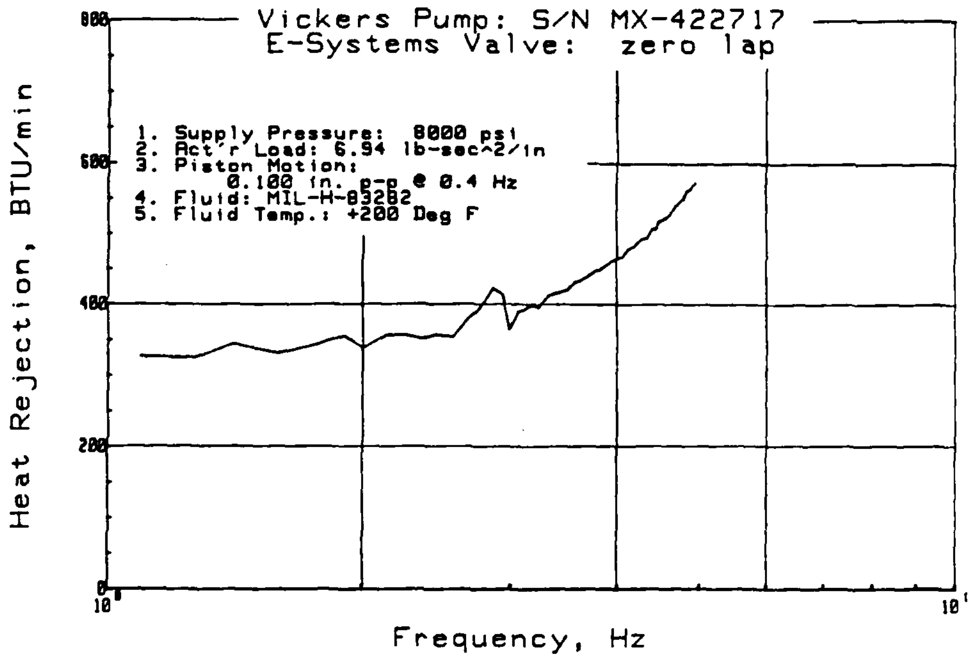

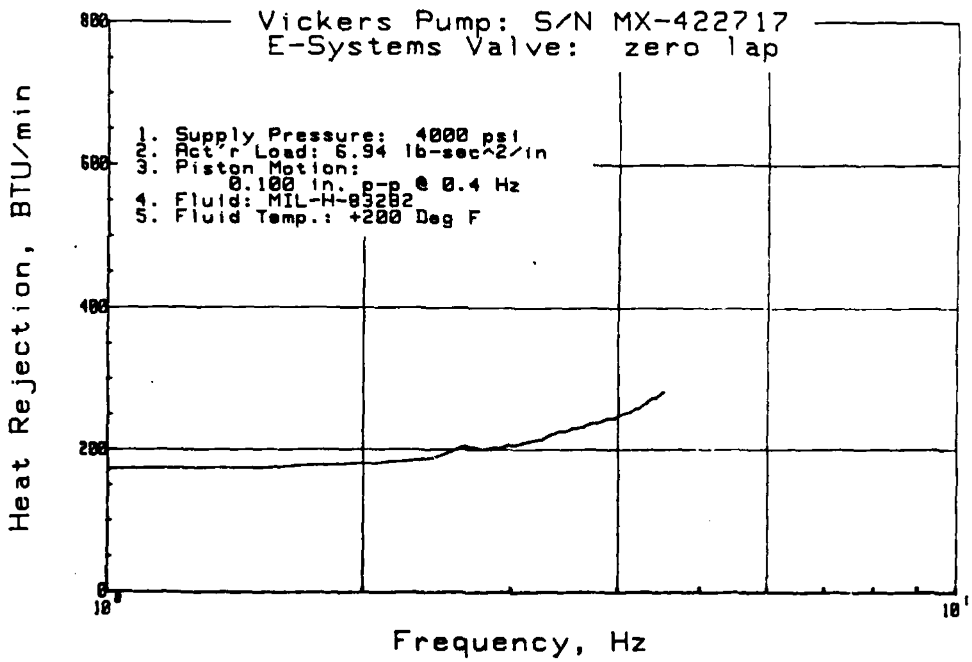

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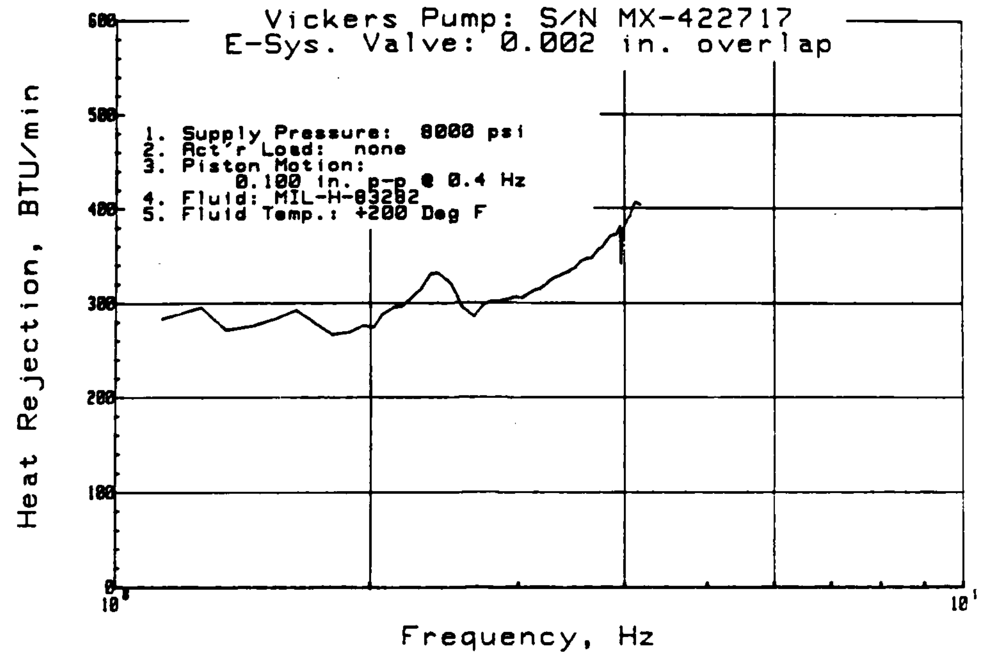

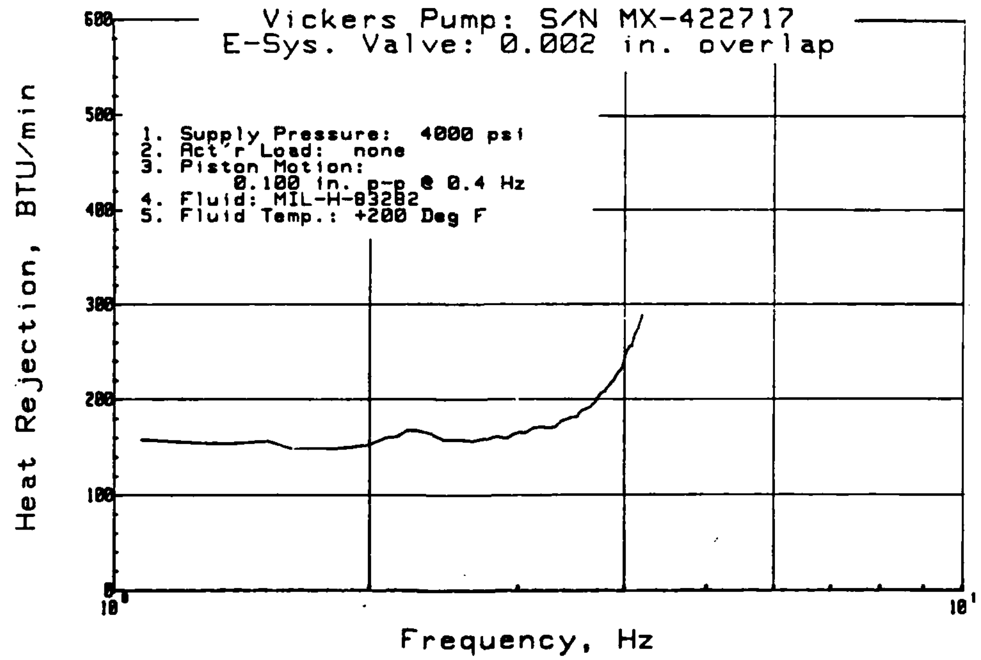

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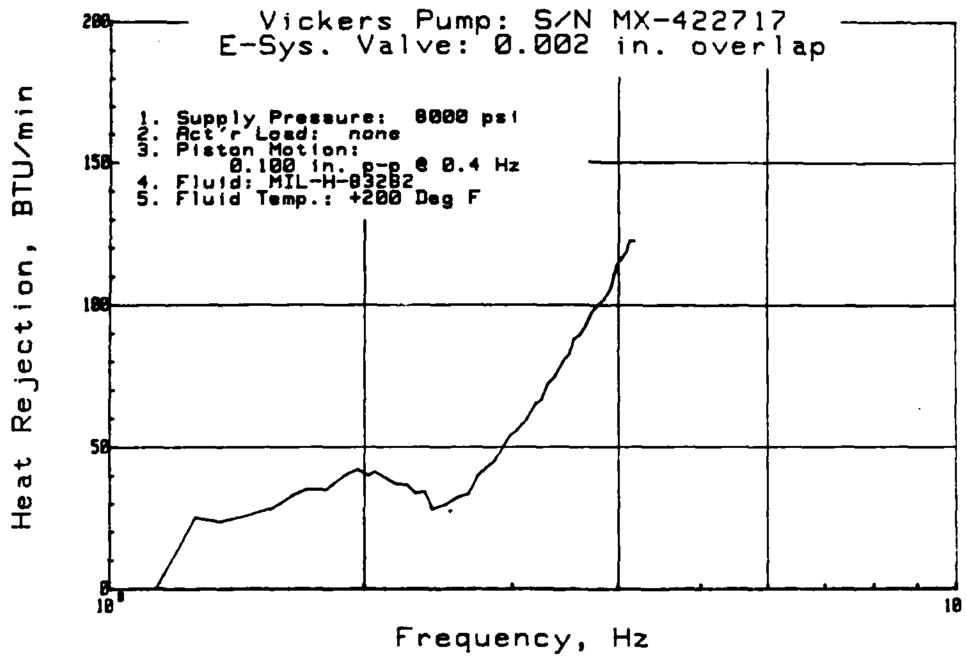

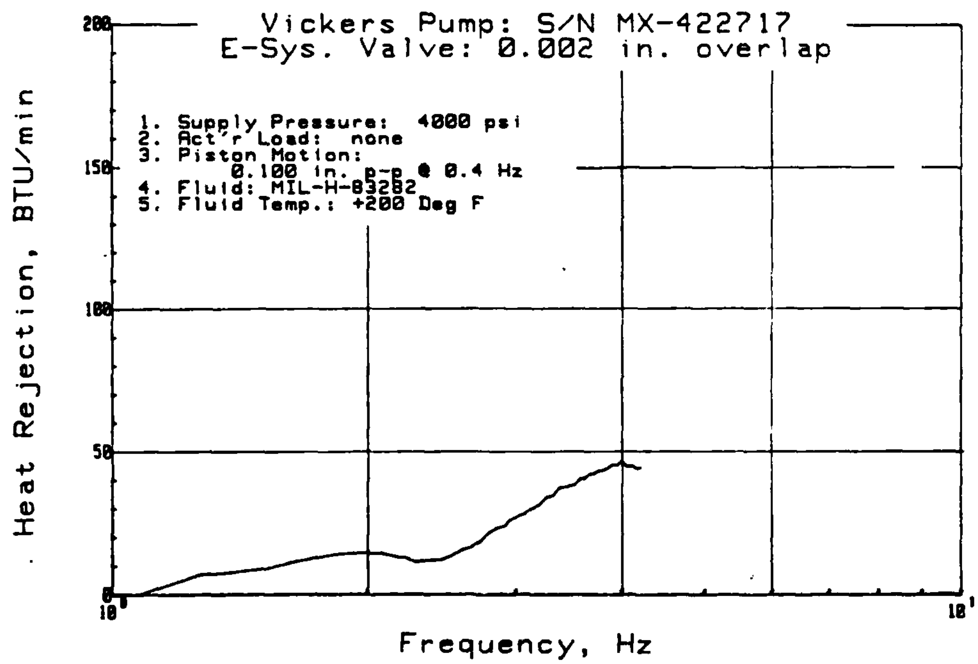

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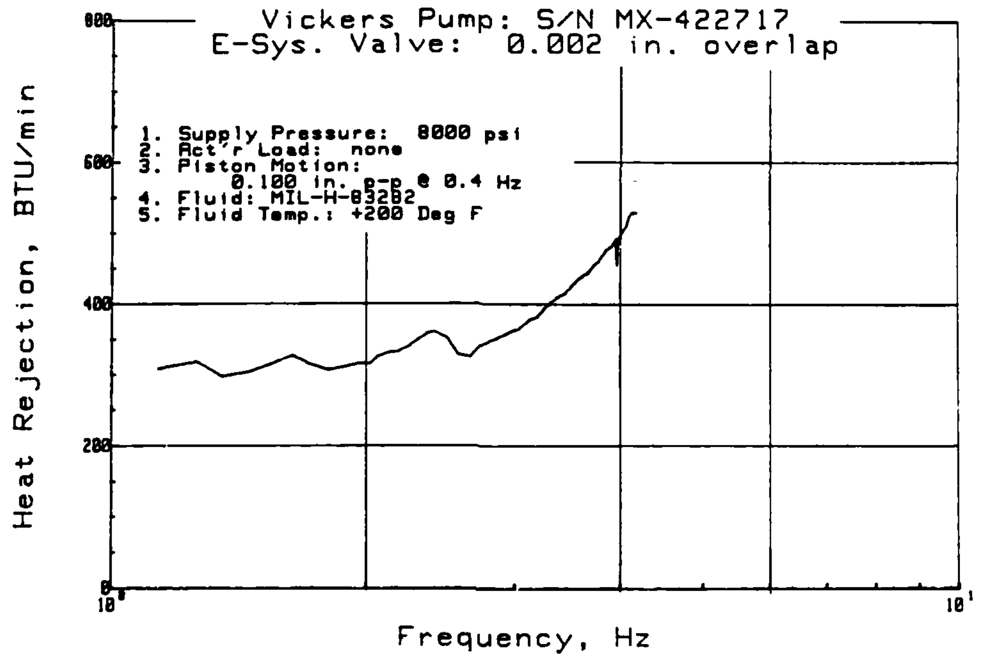

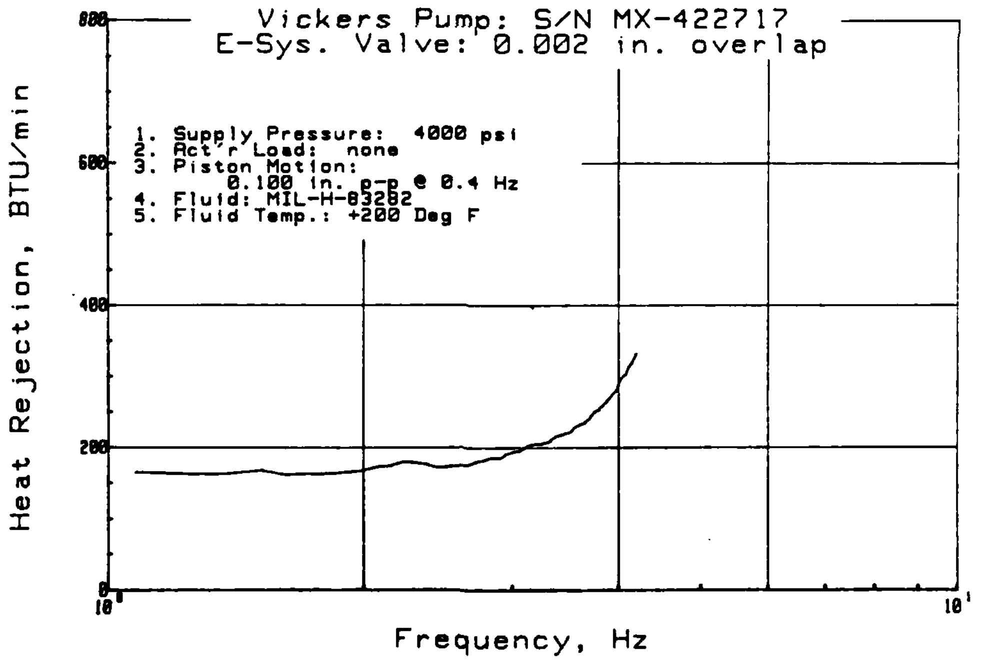

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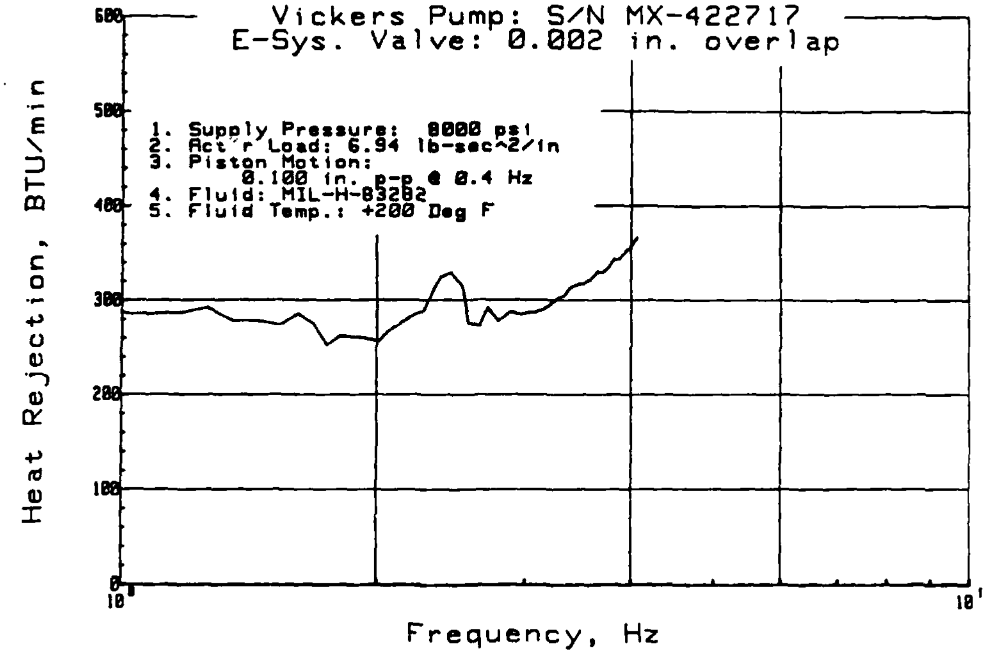

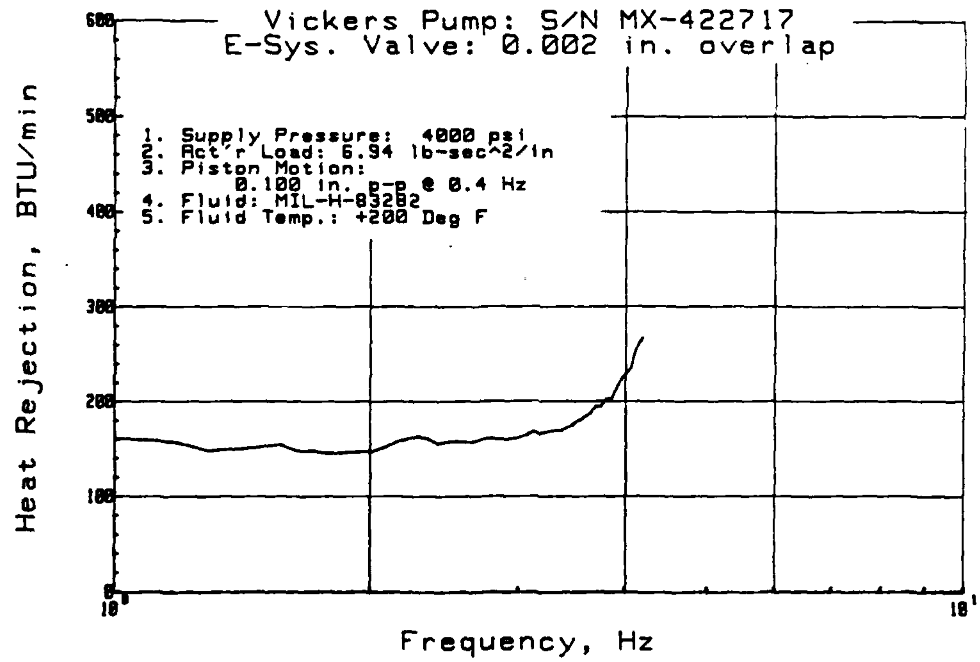

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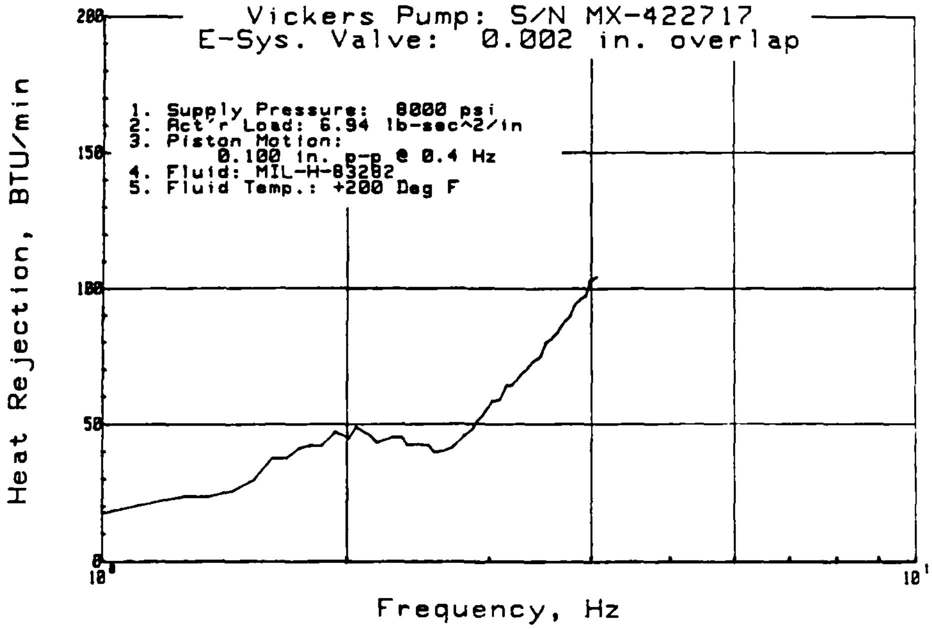

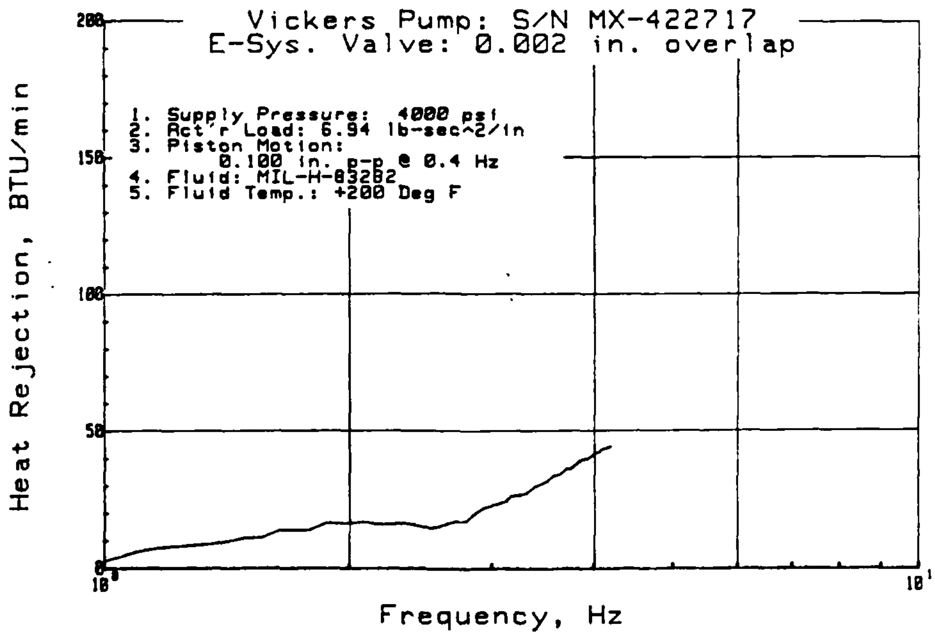

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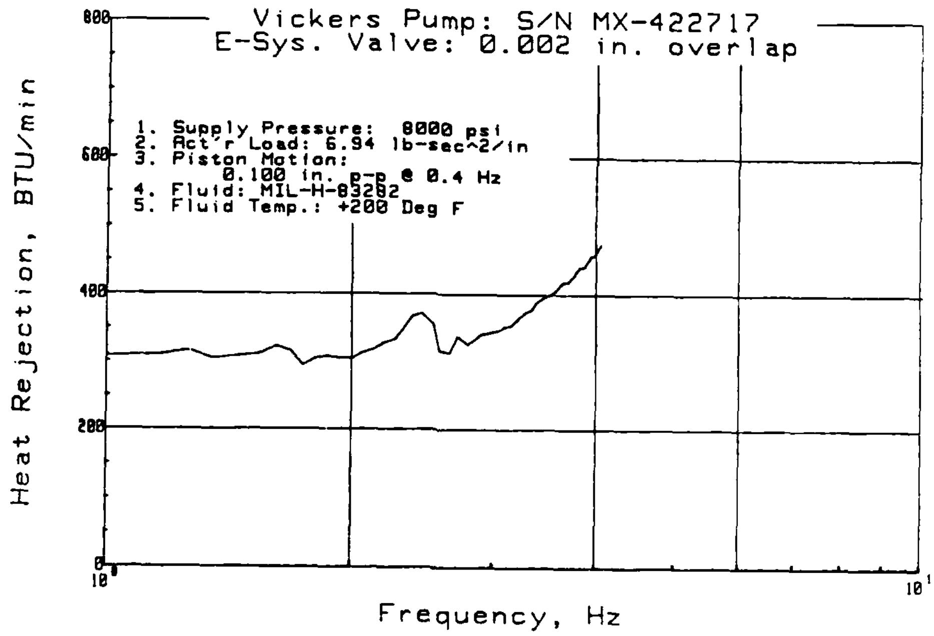
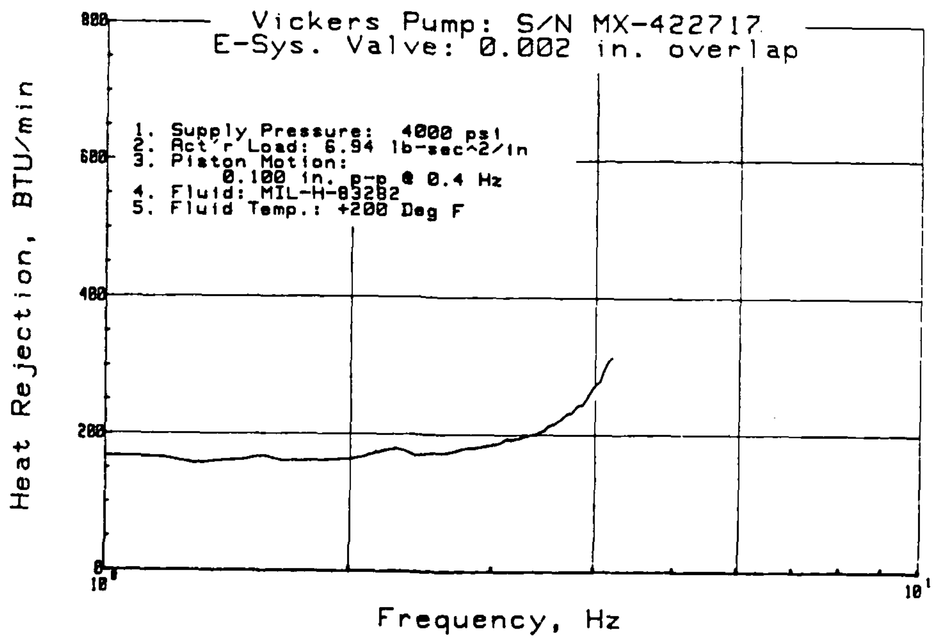




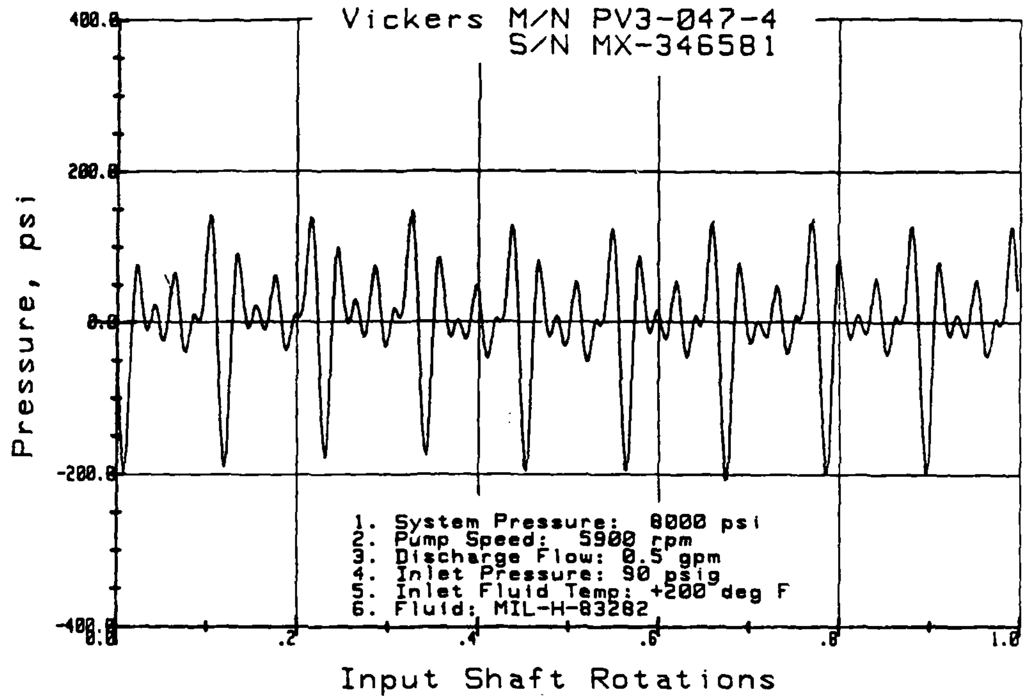
APPENDIX D

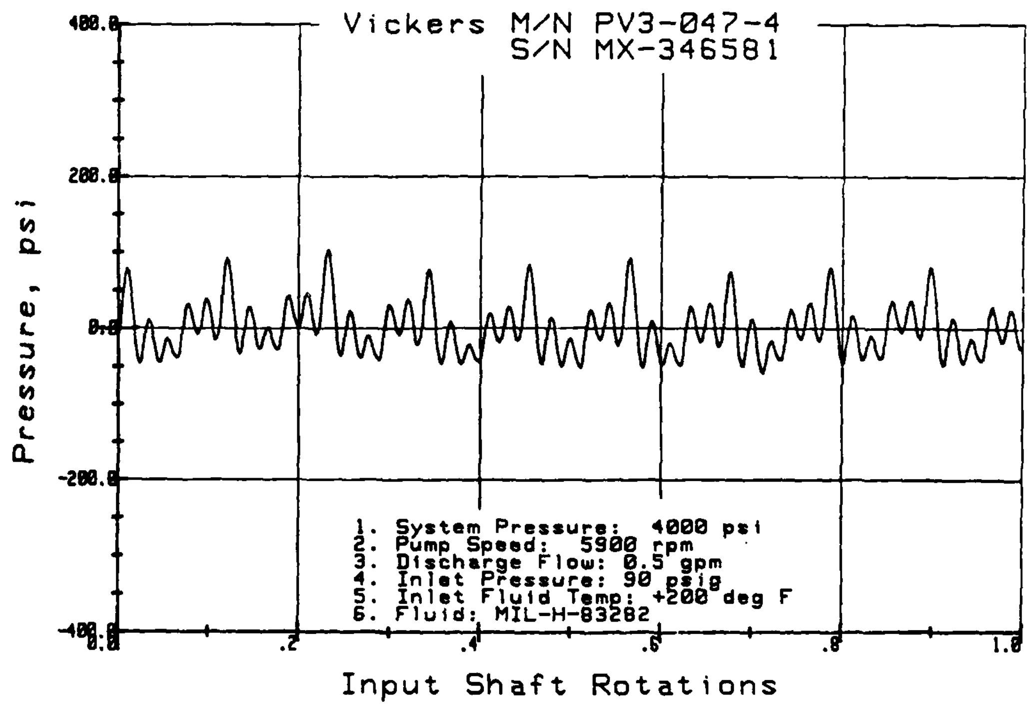
DUAL PRESSURE PUMP TEST DATA

Contents | Page

Contents	Page
Pump pressure ripple	154
Pump transient response	155
Pump performance	156
Pressure level switching at 0.5 gpm	157
Pressure level switching at 5.0 gpm	158

PUMP PRESSURE RIPPLE





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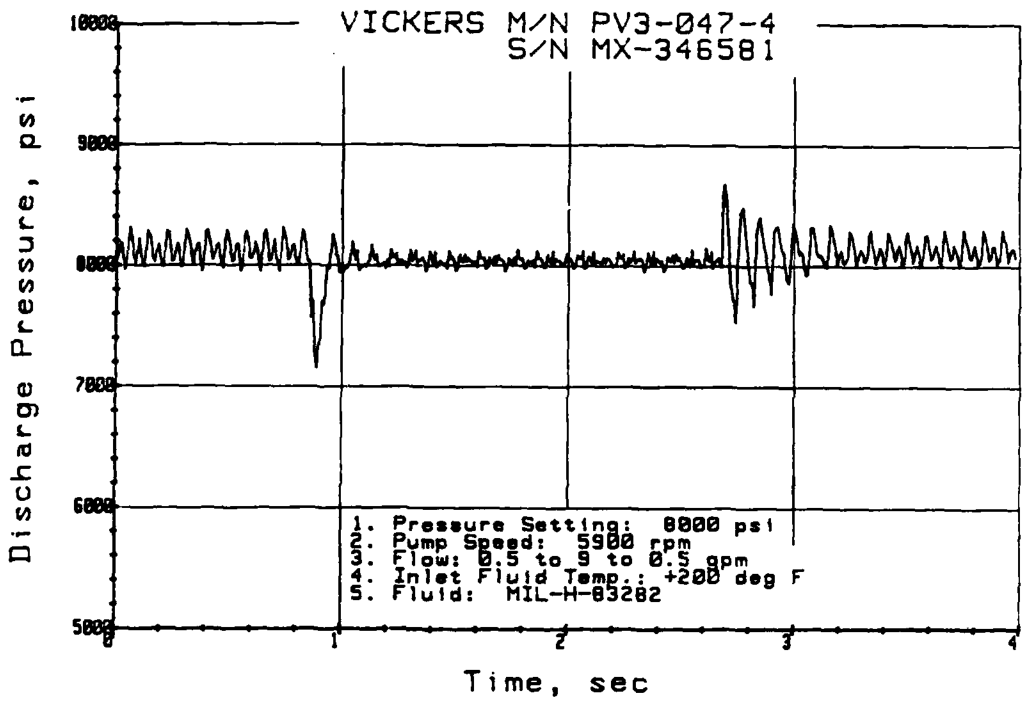

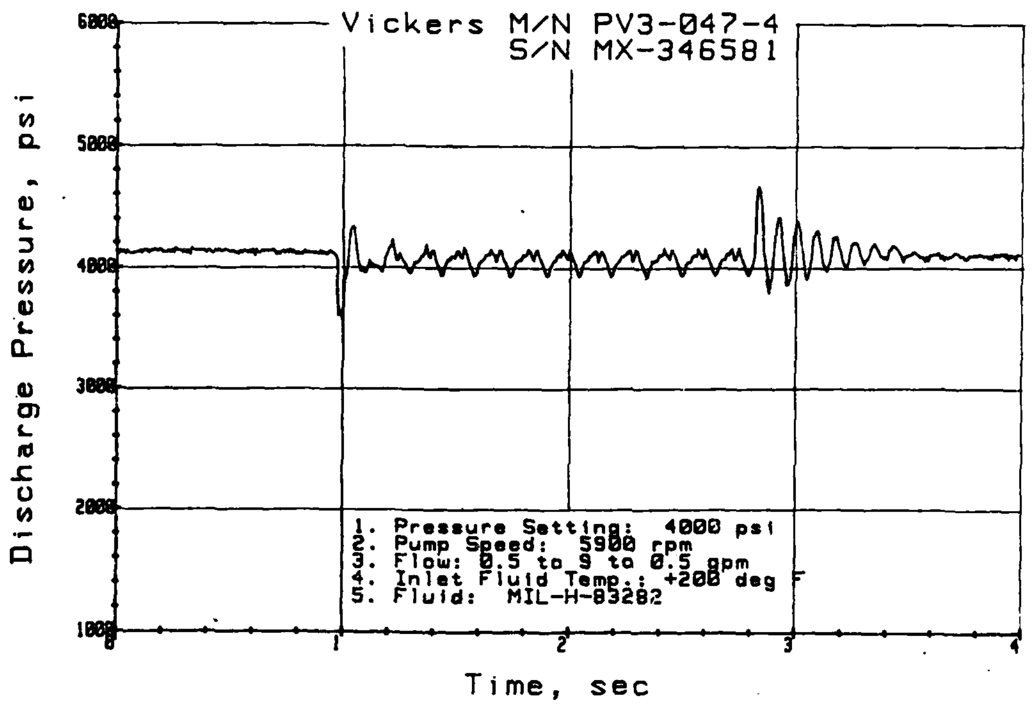

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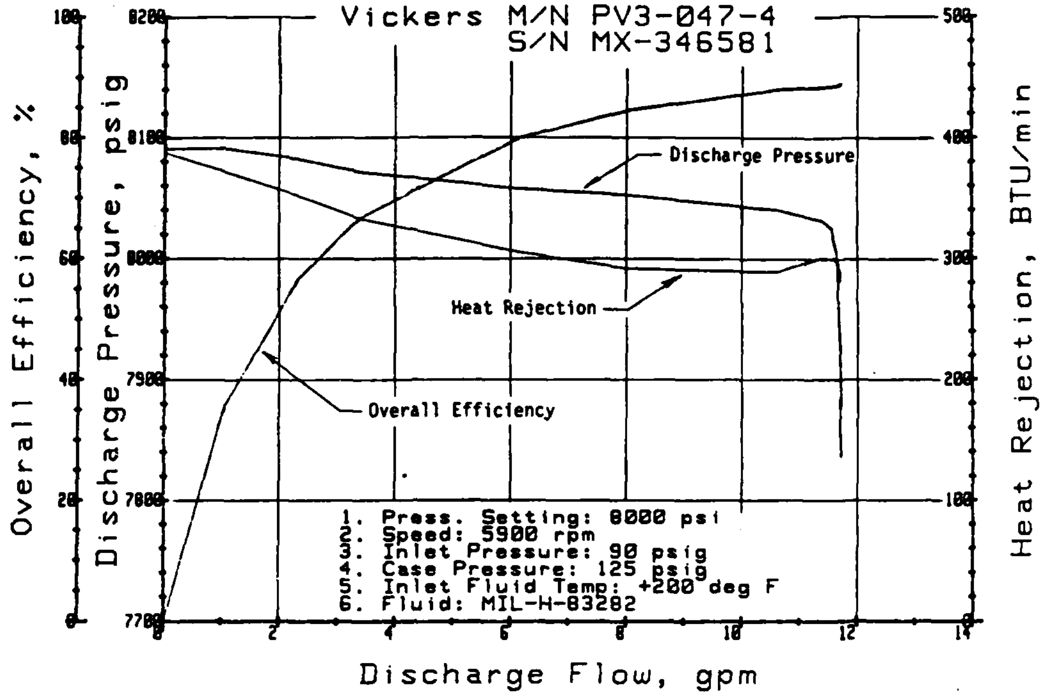

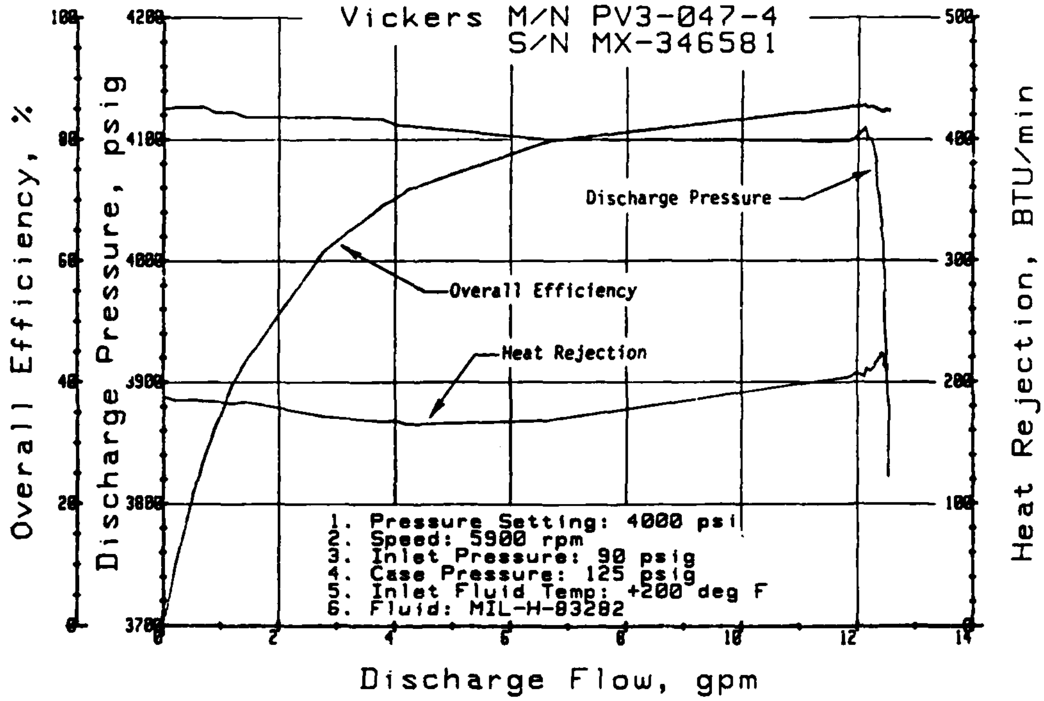

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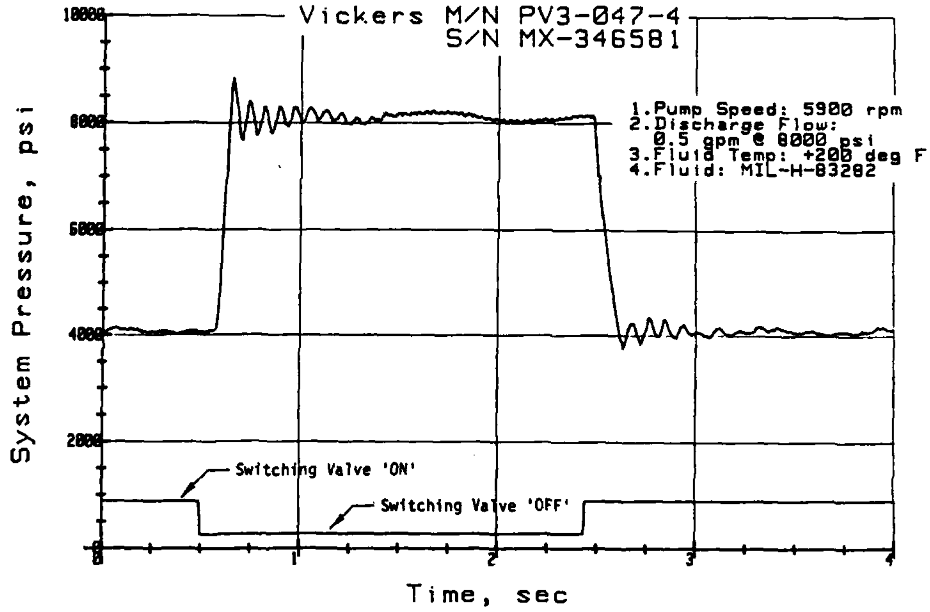
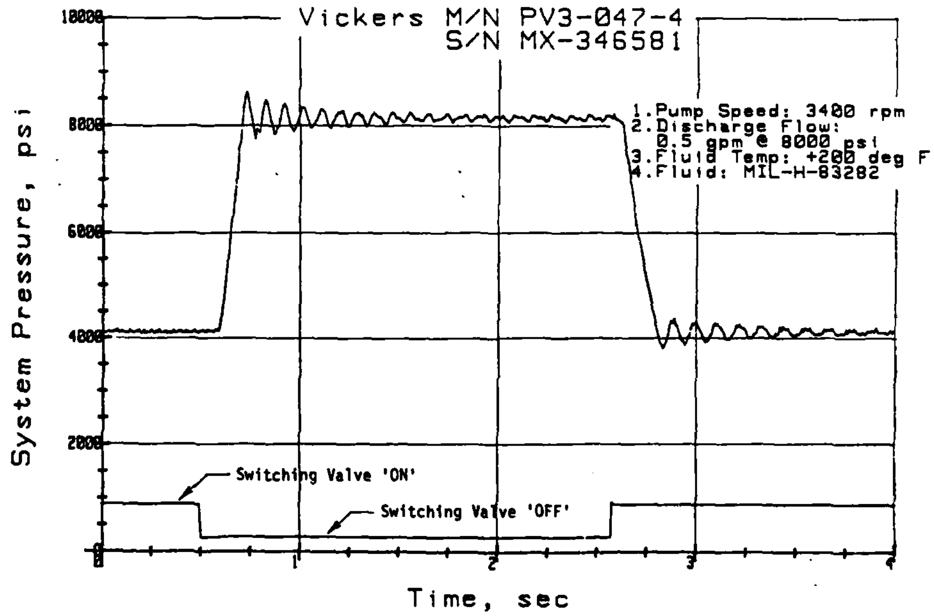
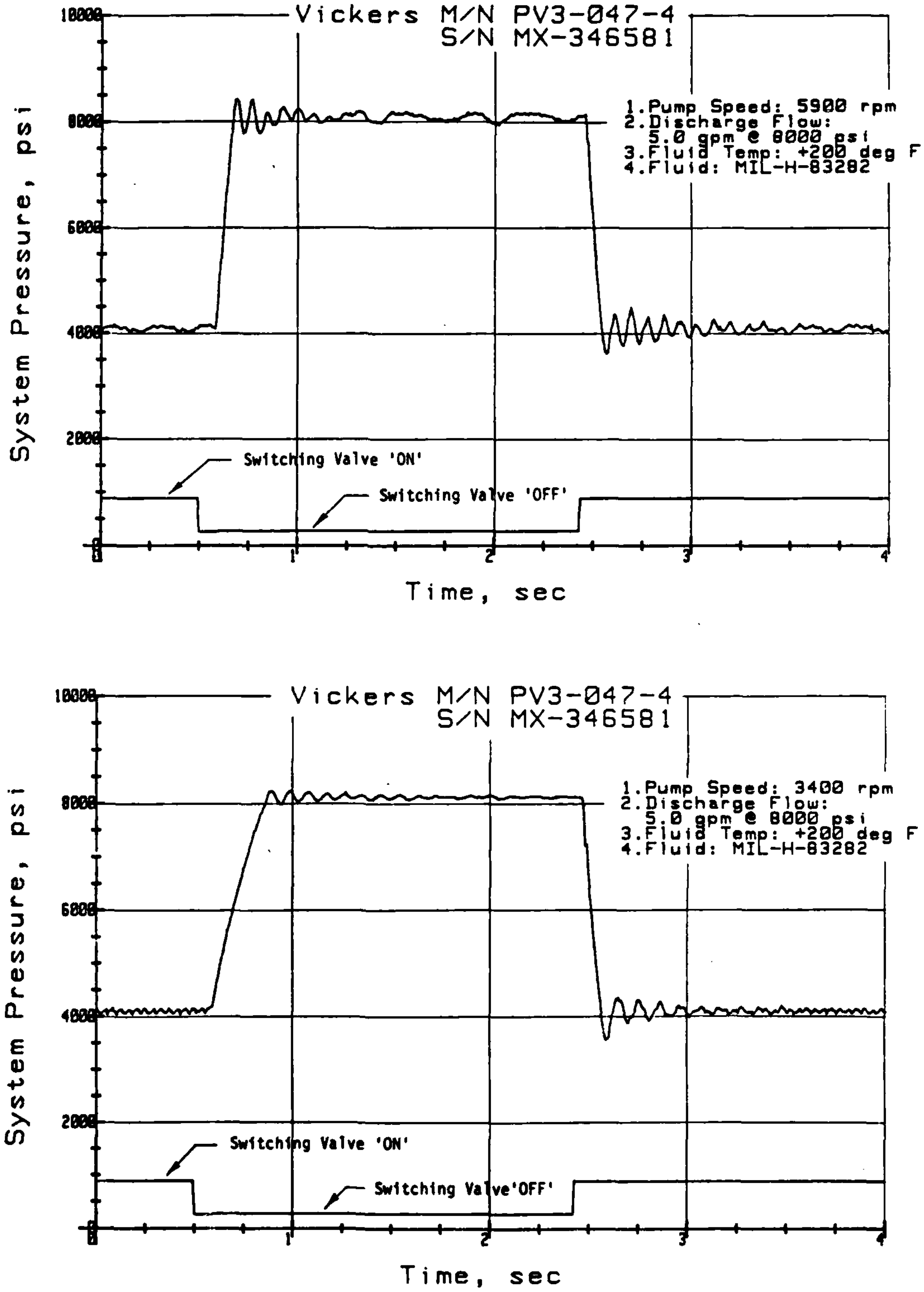




NADC-88066-60

PRESSURE LEVEL SWITCHING

PRESSURE LEVEL SWITCHING



APPENDIX E

LHS SIMULATOR TEST DATA

Contents

Contents	Page
Dynamic Response Tests	
Step response, E-Systems zero lap valve	161
Step response, E-Systems overlapped valve	162
Frequency response, E-Systems zero lap valve	163
Frequency response, E-Systems overlapped valve	164
Pressure Level Switching, Actuator Tests	
Actuator Piston Motionless, 5000 lb tension load	
E-Systems zero lap valve	165
E-Systems overlapped valve	166
Actuator Piston Moving, 4000 to 6000 lb tension load	
E-Systems zero lap valve, 0° switching point	167
E-Systems zero lap valve, 90° switching point	168
E-Systems overlapped valve, 0° switching point	169
E-Systems overlapped valve, 90° switching point	170
Pressure Level Switching, System Tests	
Actuator operating mode, 2% load/stroke	171
Actuator operating mode, 10% load/stroke	172
Actuator operating mode, 50% load/stroke	173
Pressure Ripple Tests	
FC-1 System	174
FC-2 System	175
Spectrum Analysis	
FC-1 System, 4000 psi, low pump speeds	176
FC-1 System, 4000 psi, high pump speeds	177
FC-1 System, 8000 psi, low pump speeds	178
FC-1 System, 8000 psi, high pump speeds	179
FC-2 System, 4000 psi, low pump speeds	180
FC-2 System, 4000 psi, high pump speeds	181
FC-2 System, 8000 psi, low pump speeds	182
FC-2 System, 8000 psi, high pump speeds	183

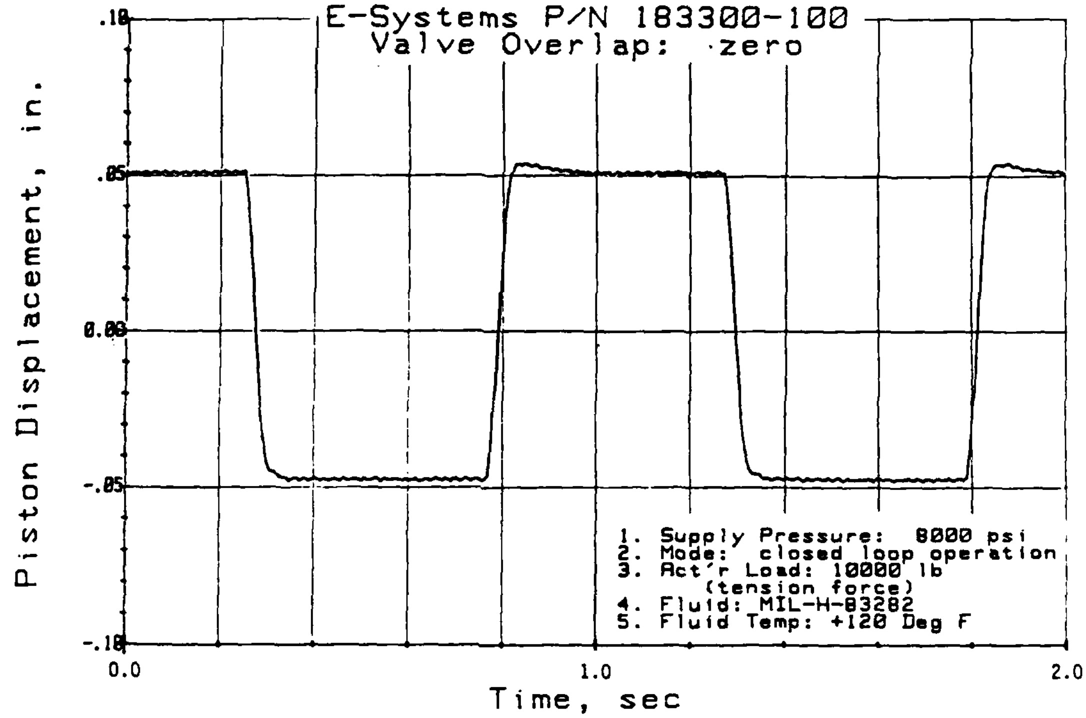
APPENDIX E

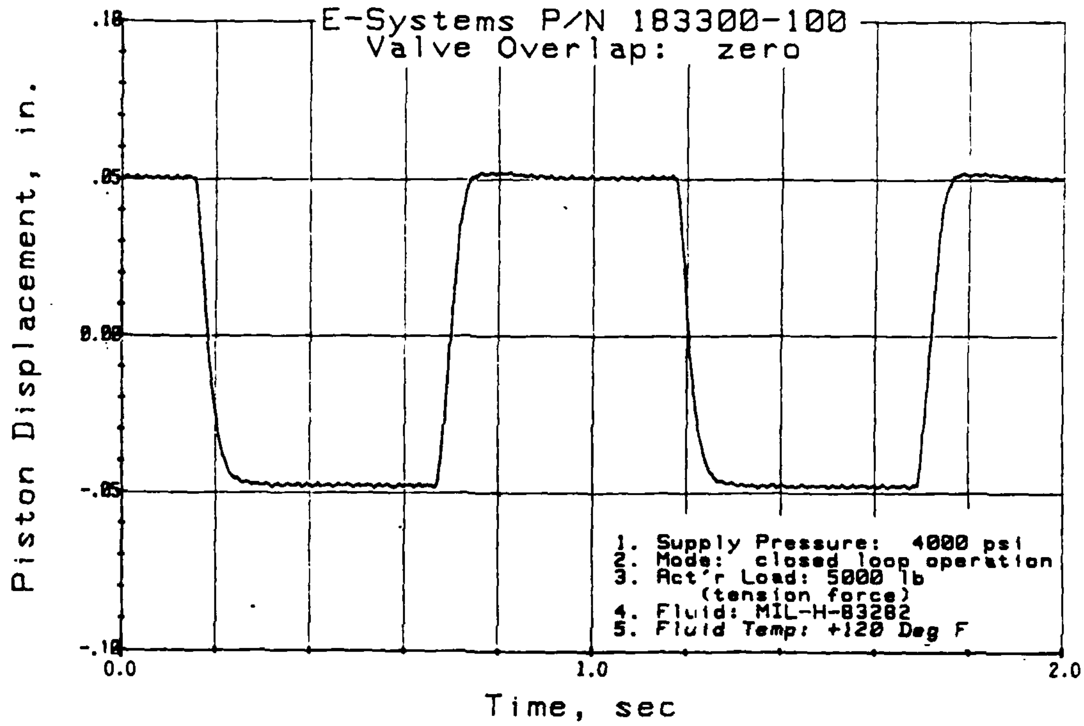
LHS SIMULATOR TEST DATA (Continued)

Contents

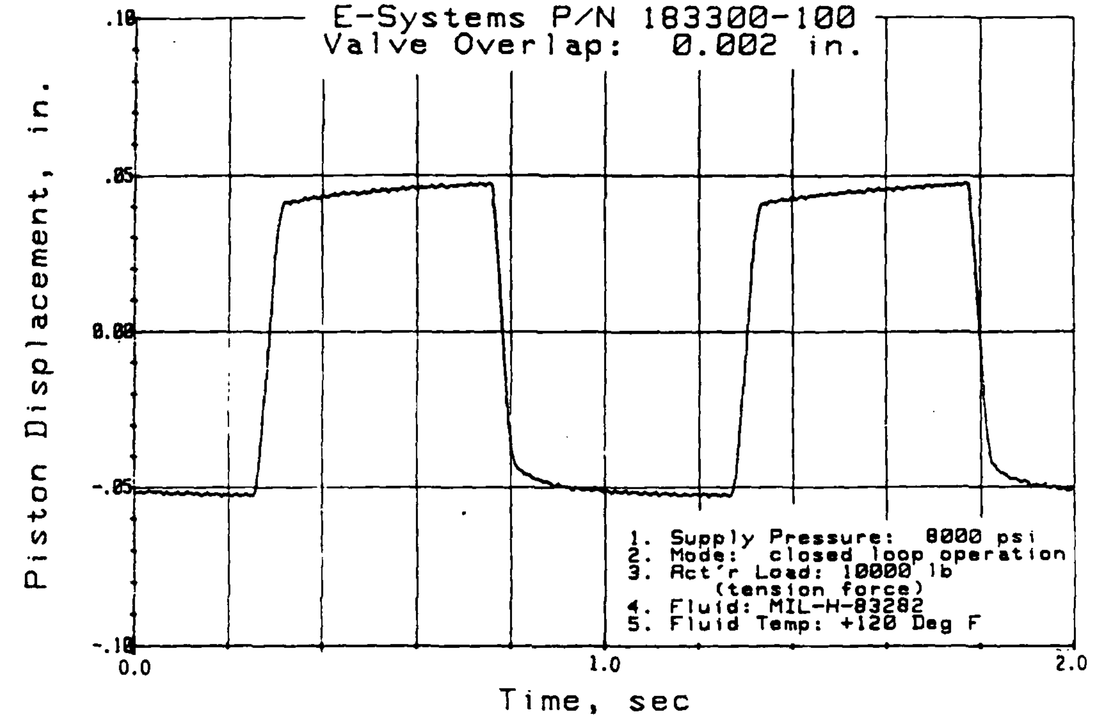
	Page
Energy Consumption Tests	
E-Systems zero lap valve, no load	
Pump	184
Actuator	185
System	186
E-Systems overlapped valve, no load	
Pump	187
Actuator	188
System	189
E-Systems zero lap valve, 5000 lb tension load	
Pump	190
Actuator	191
System	192
E-Systems overlapped valve, 5000 lb tension load	
Pump	193
Actuator	194
System	195

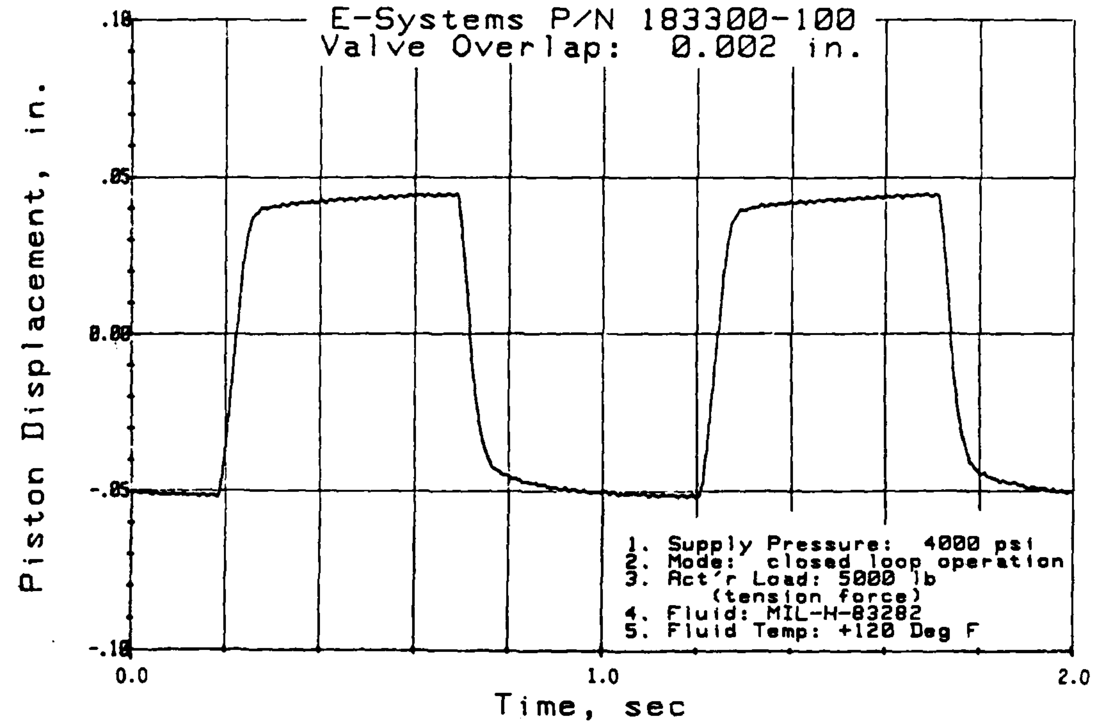
STEP RESPONSE





STEP RESPONSE





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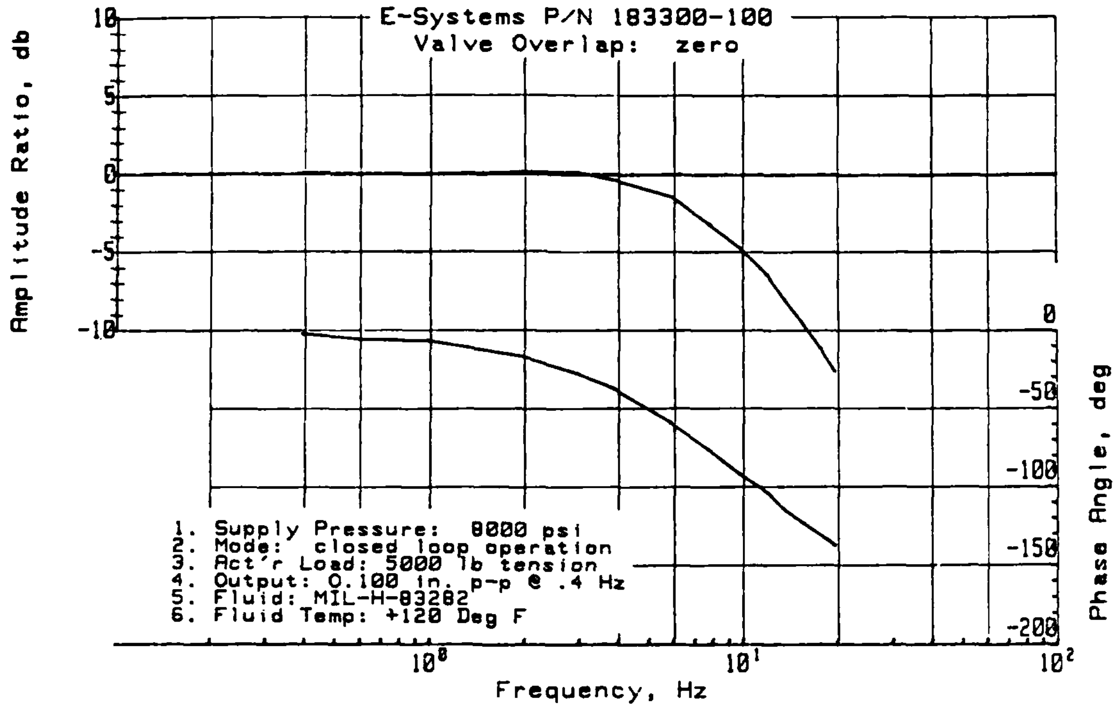

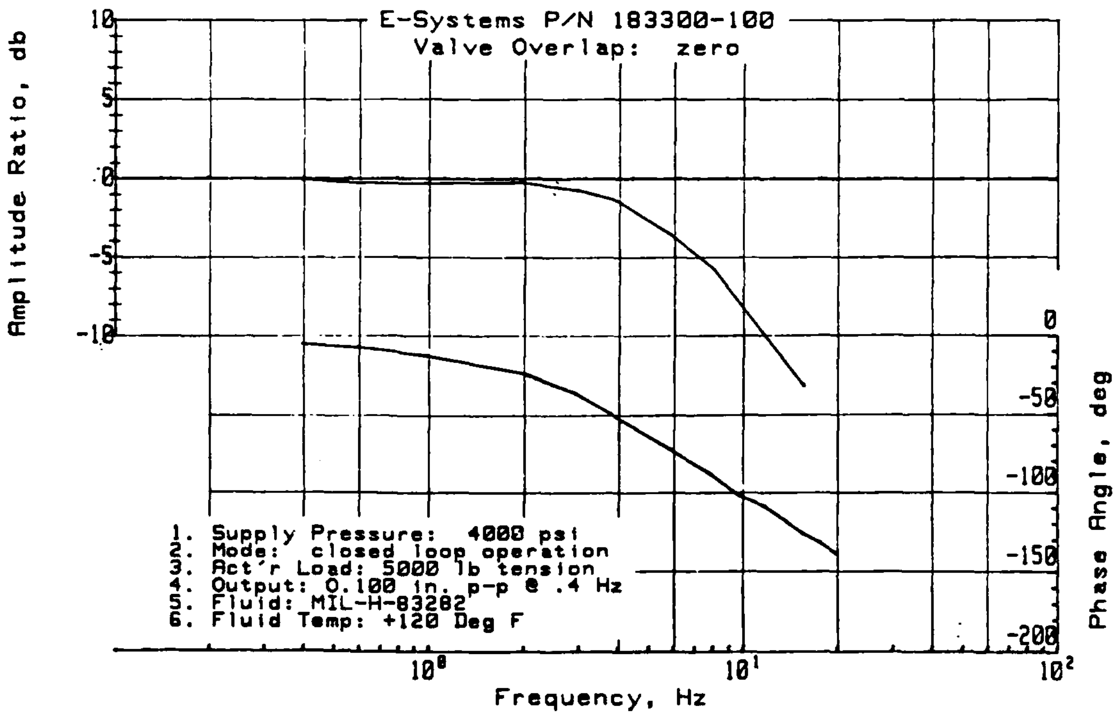

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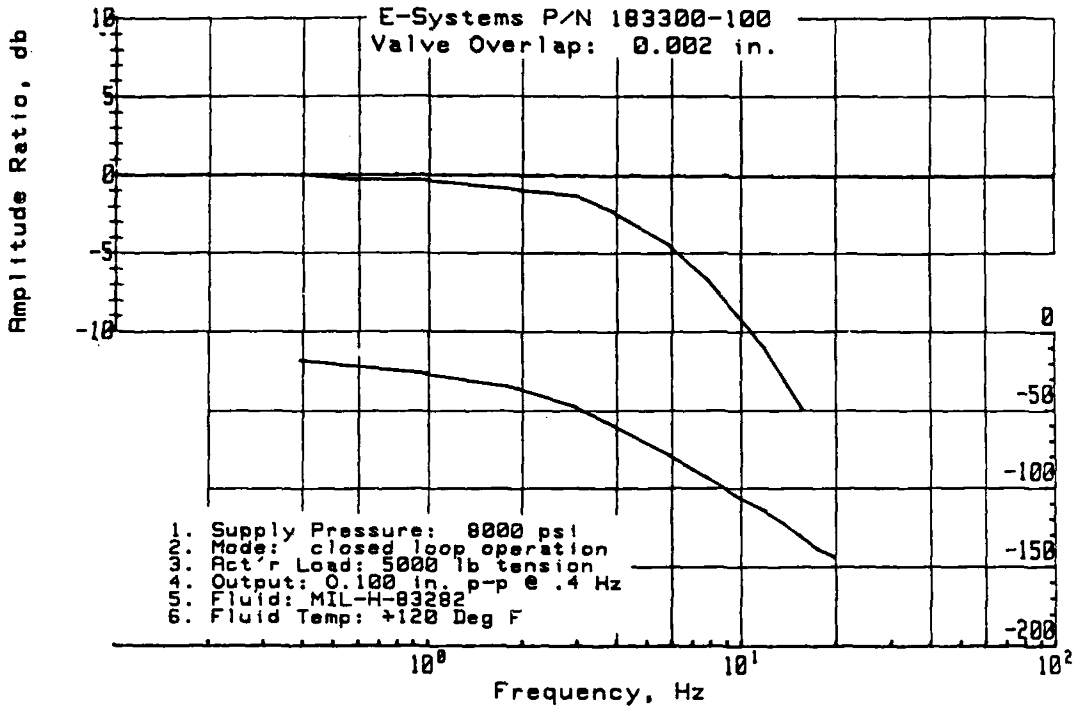

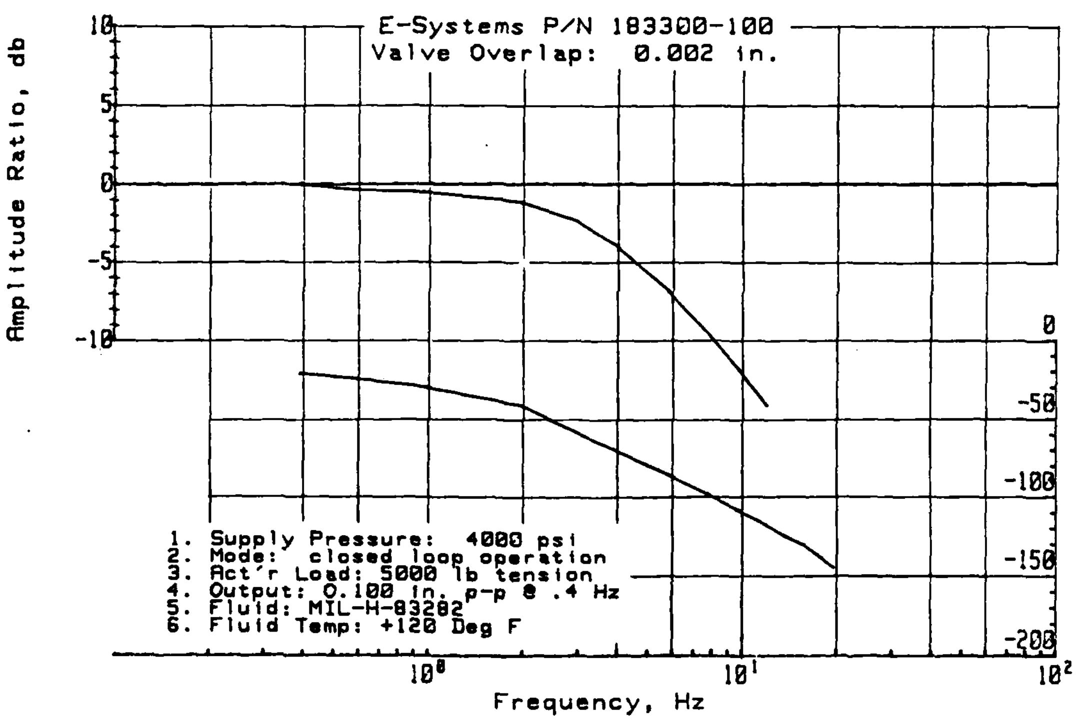

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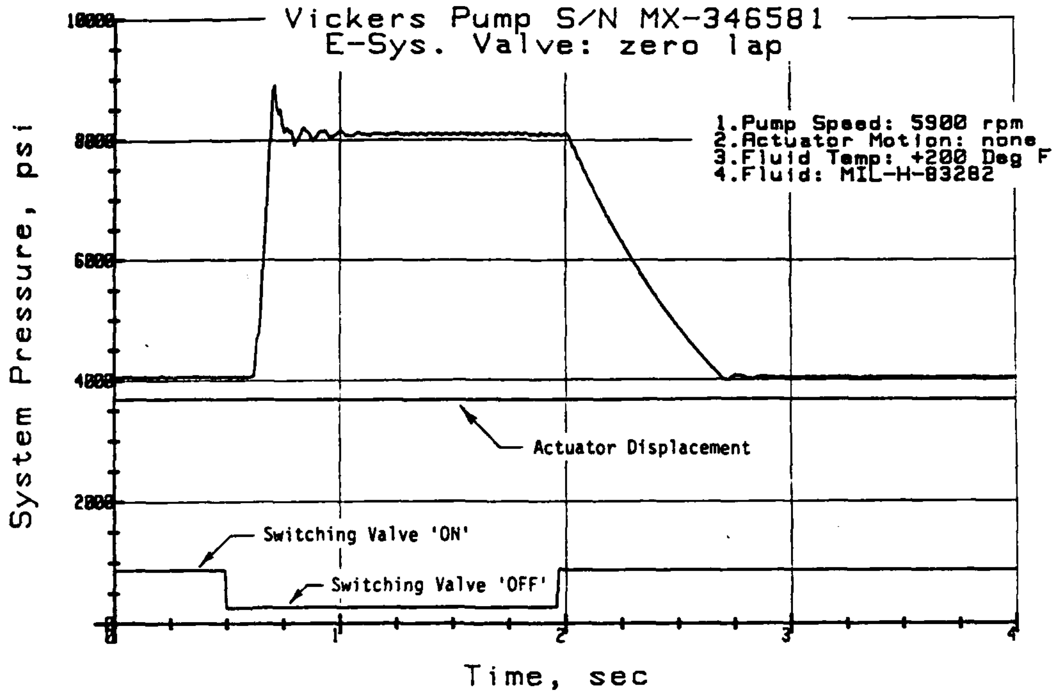

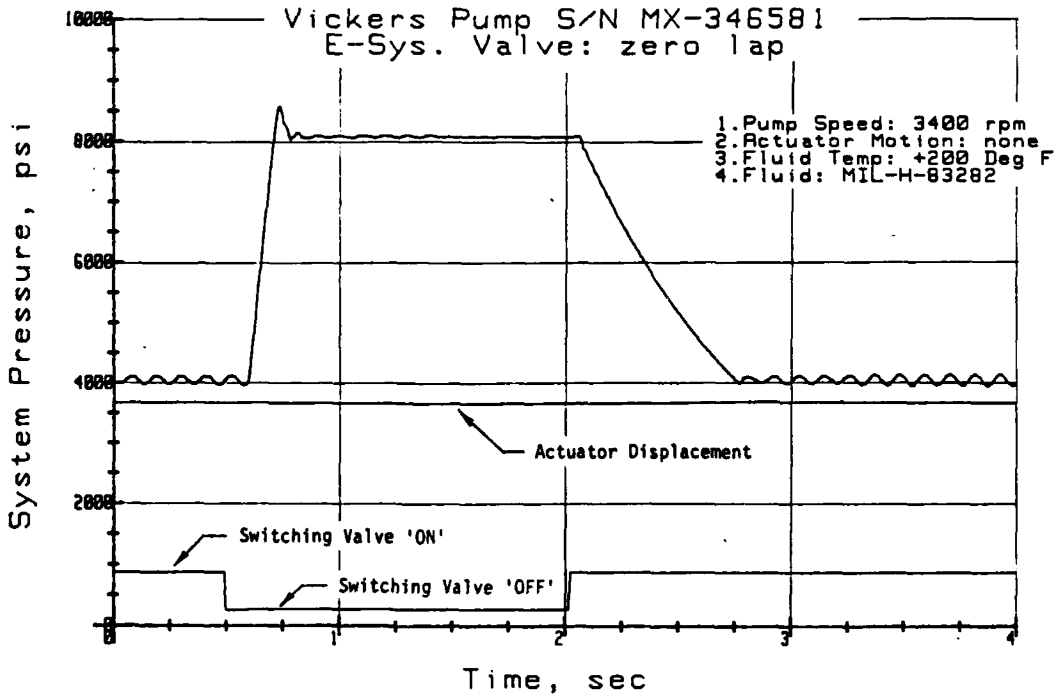

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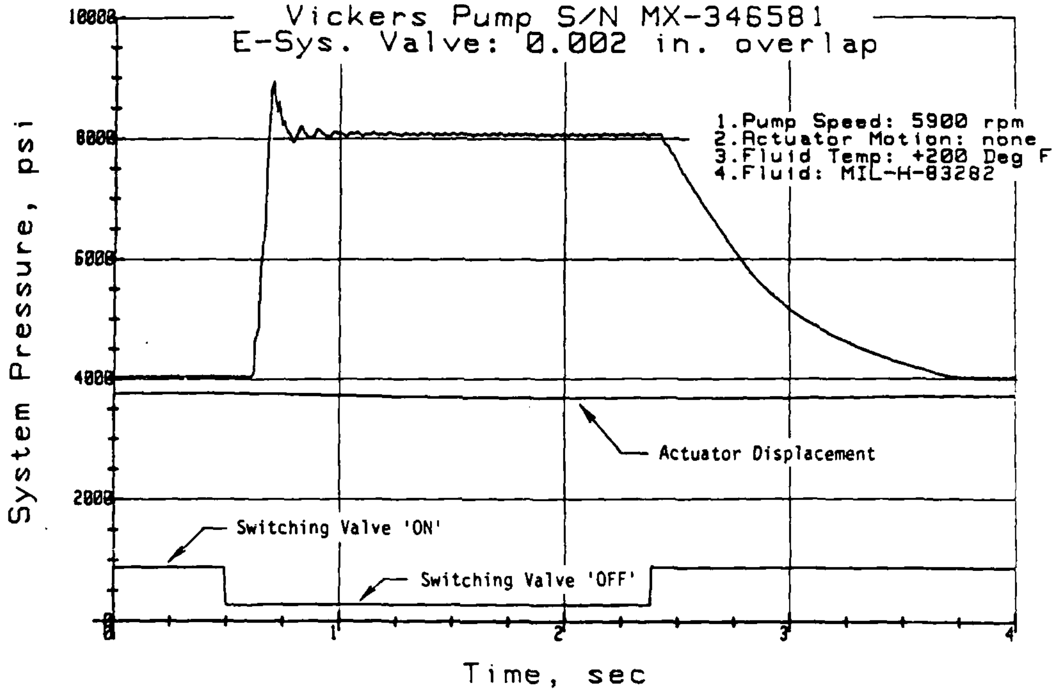

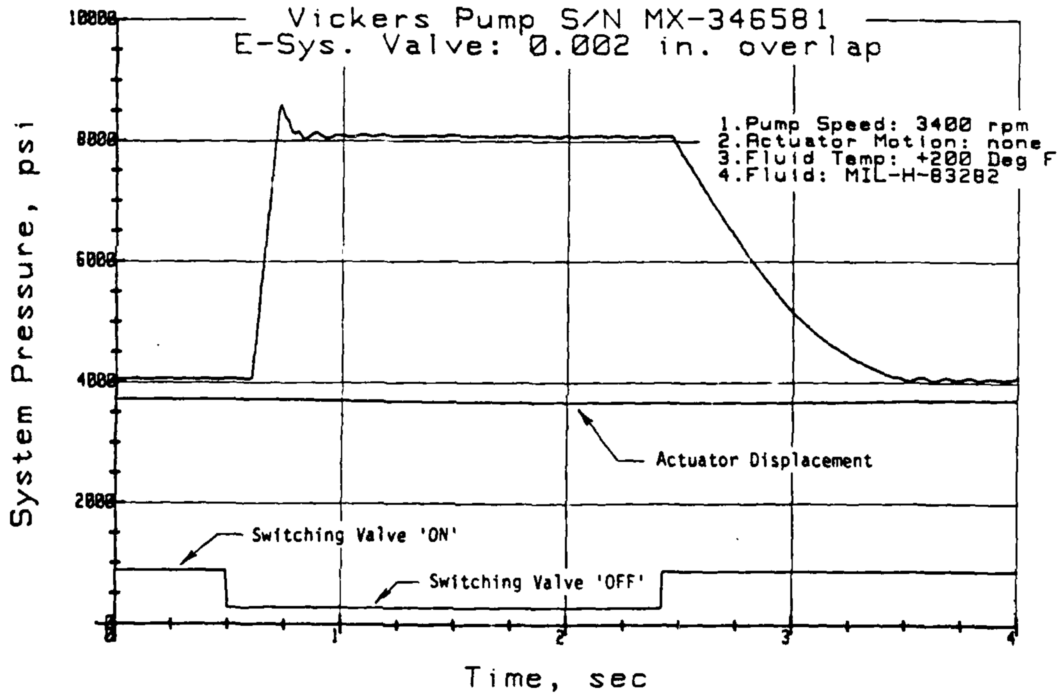

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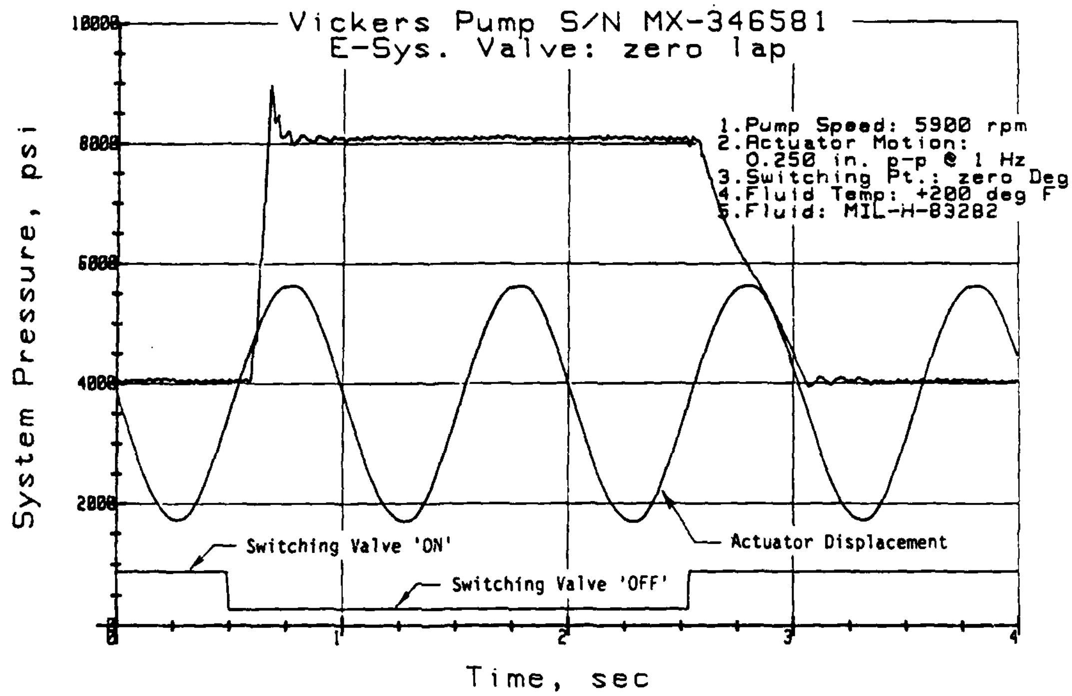

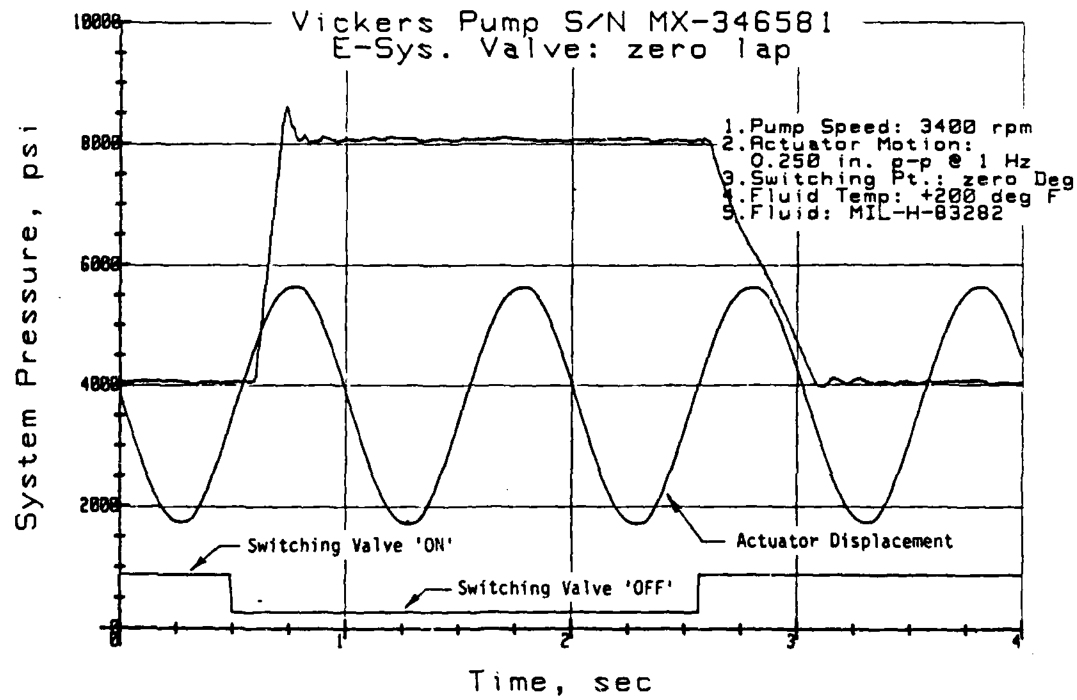

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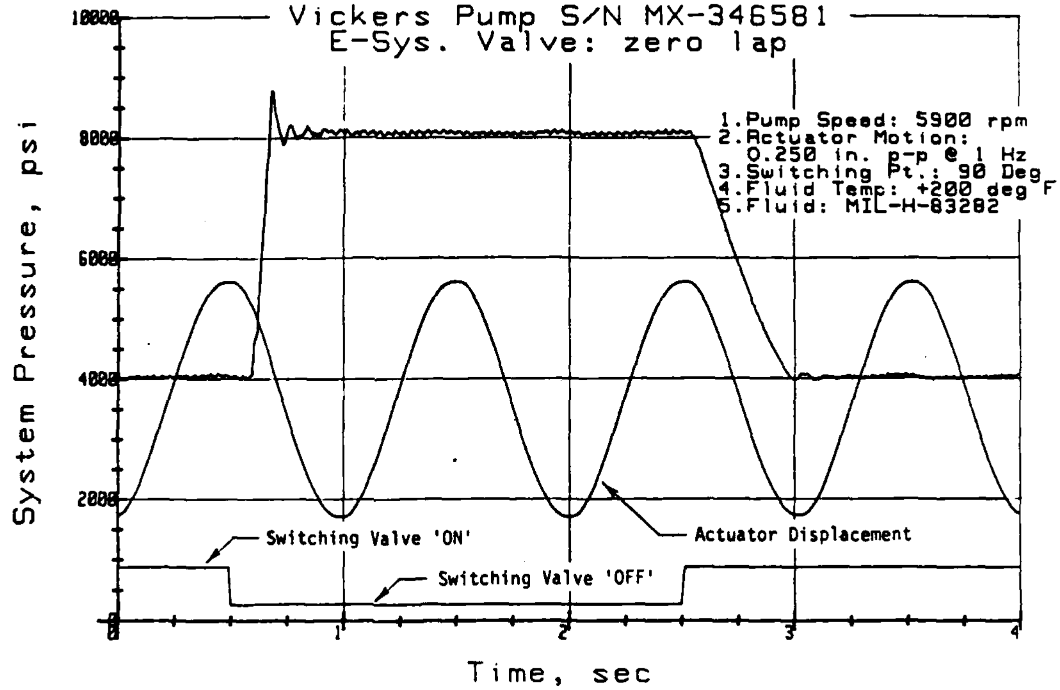

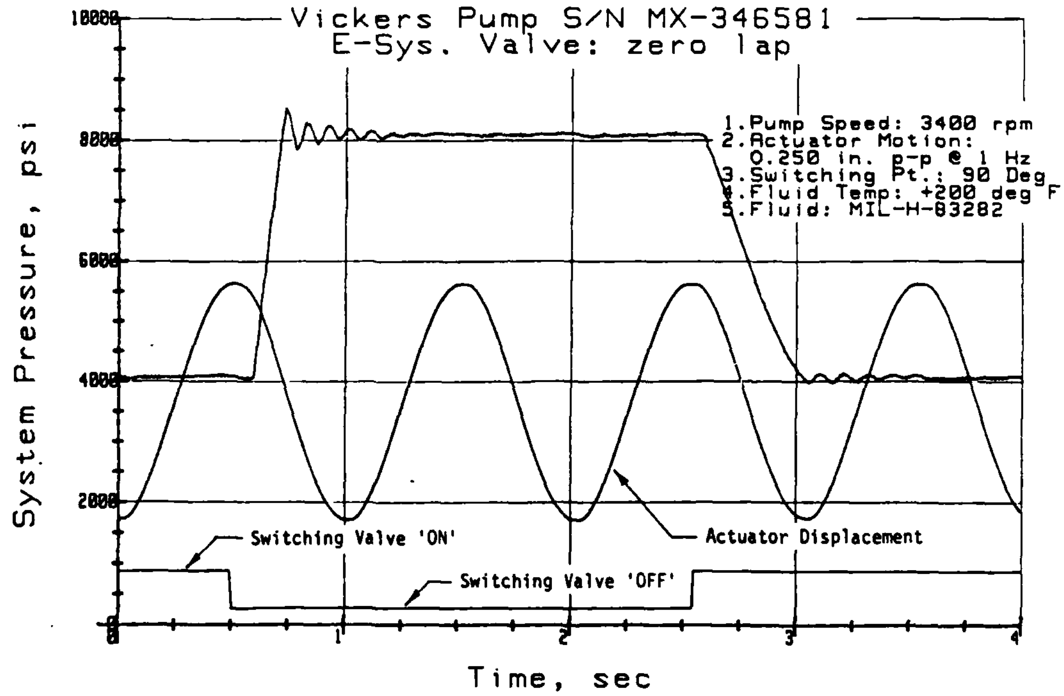

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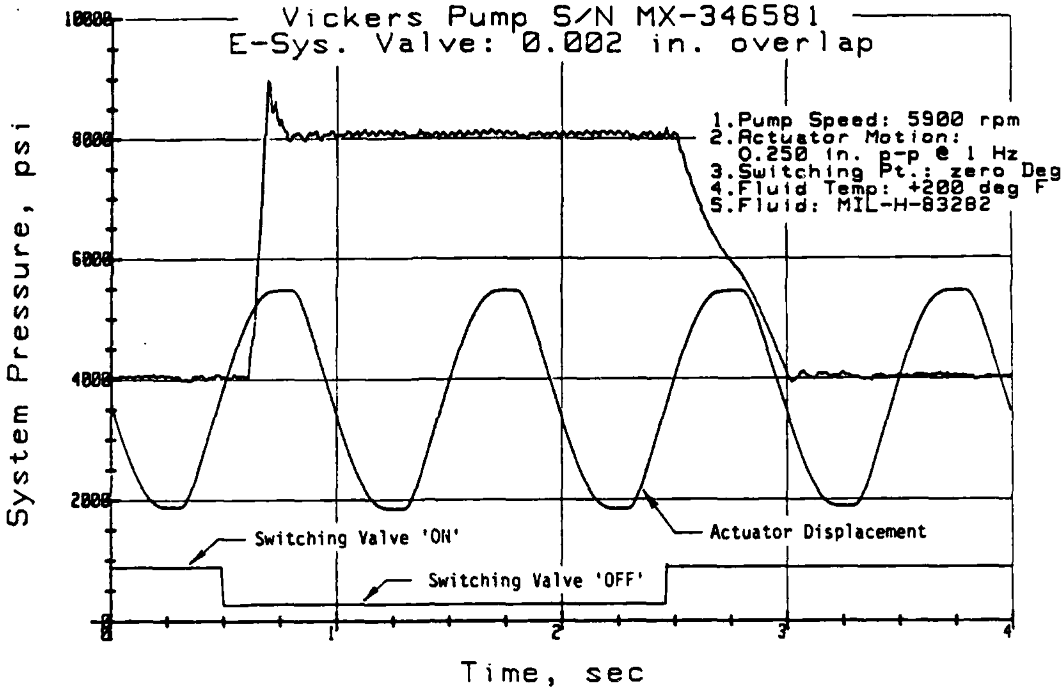

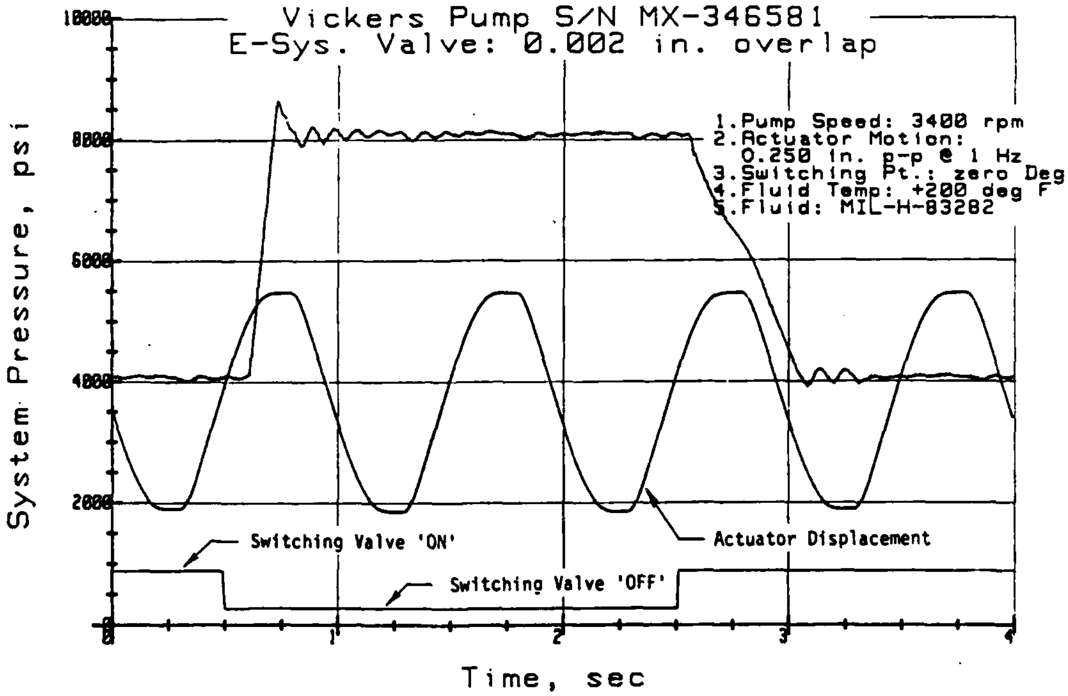

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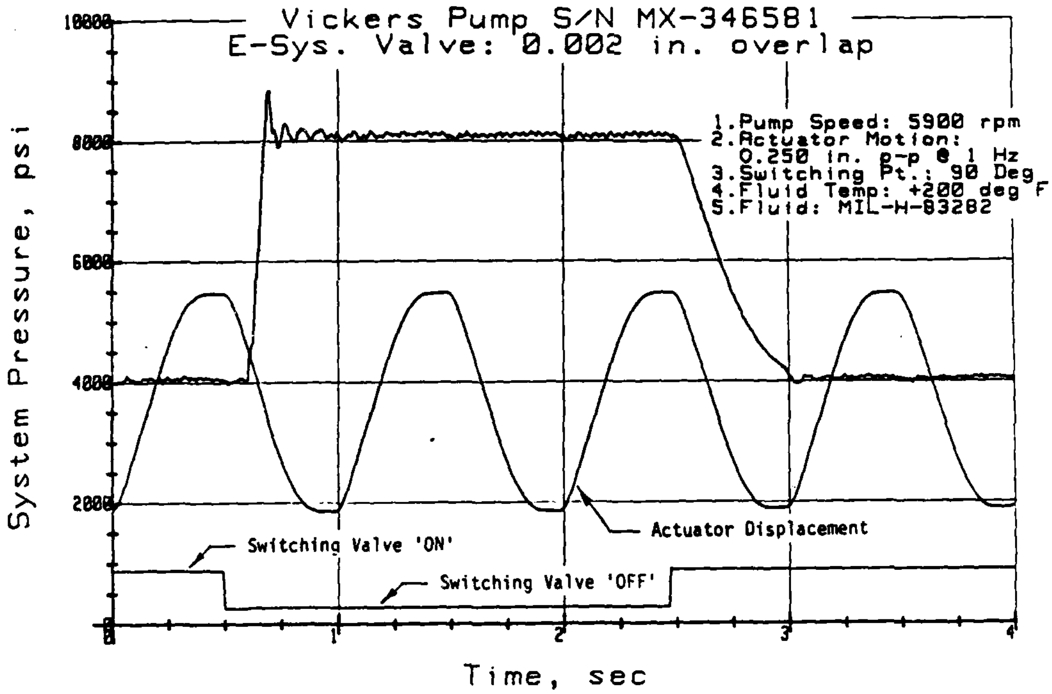

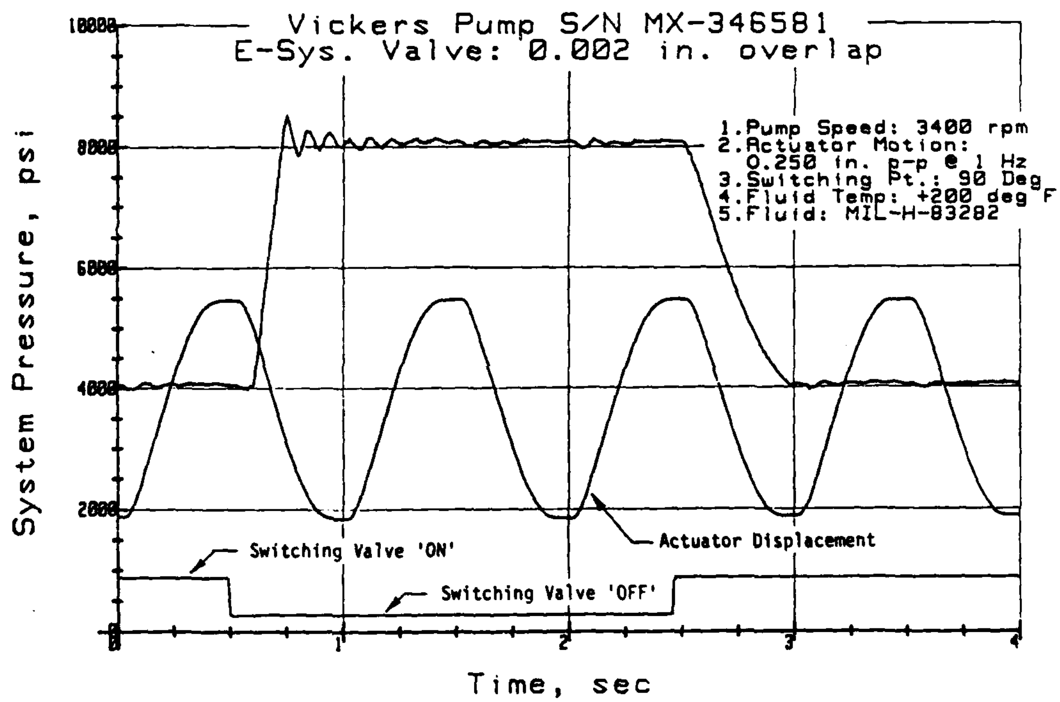

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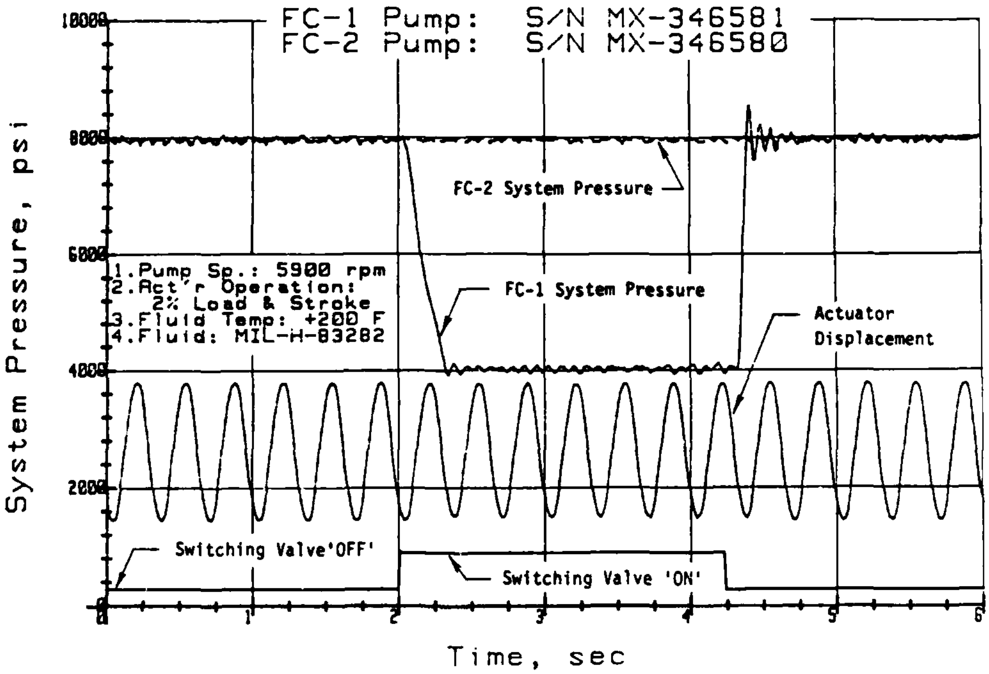

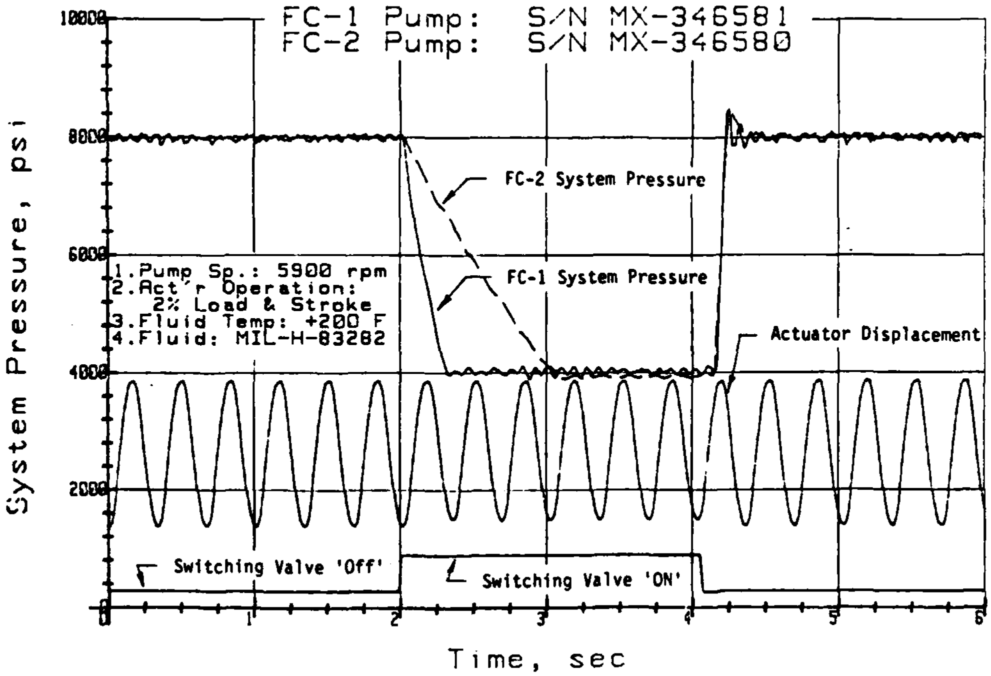

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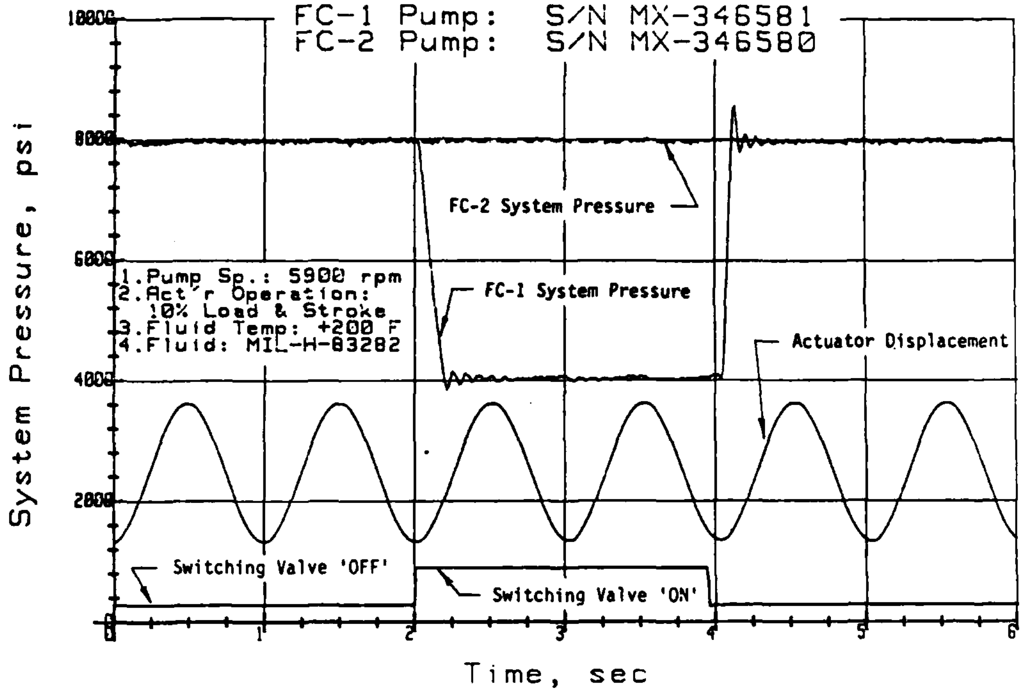

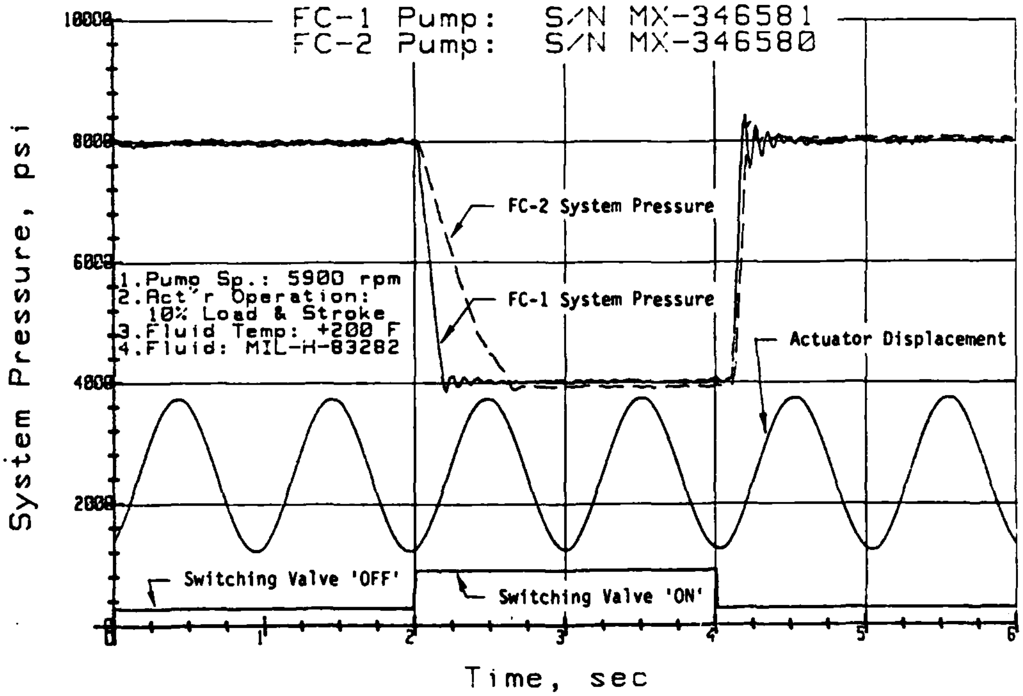

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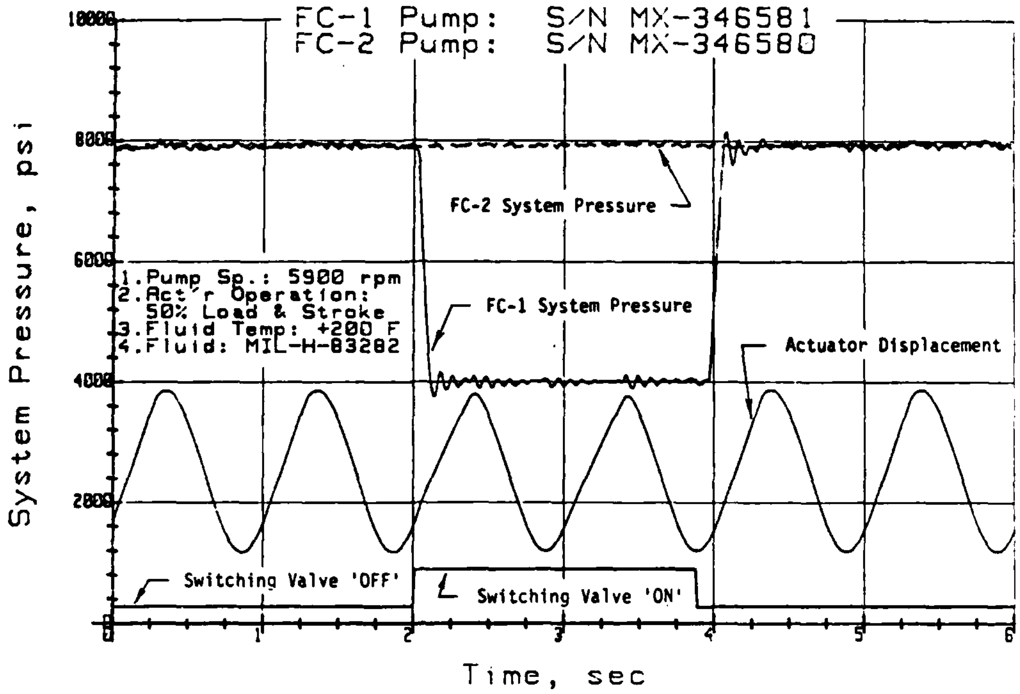
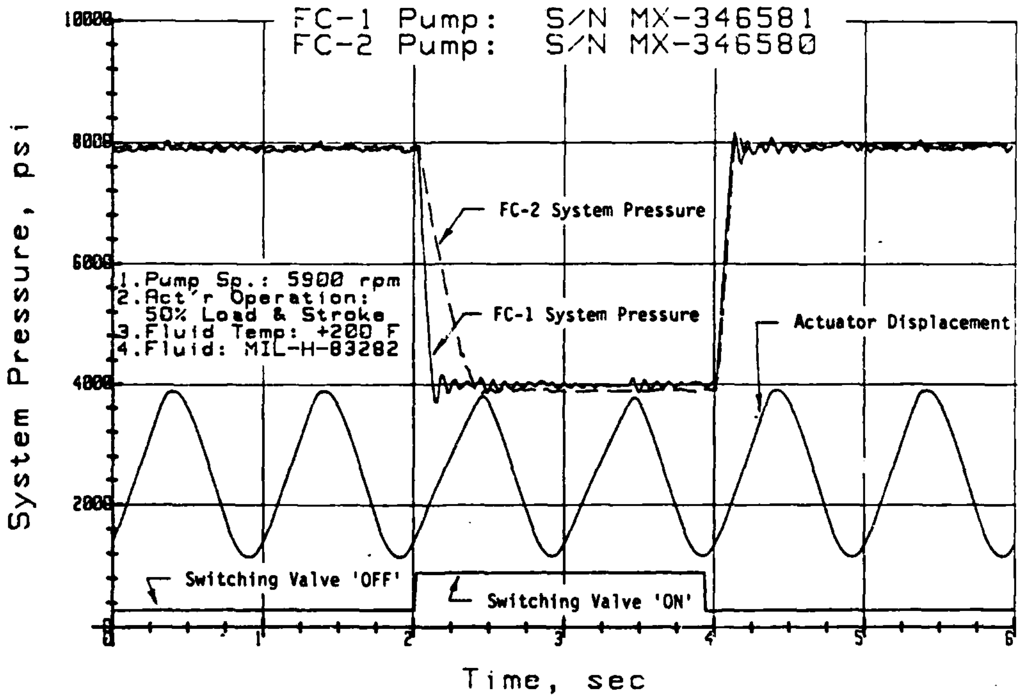
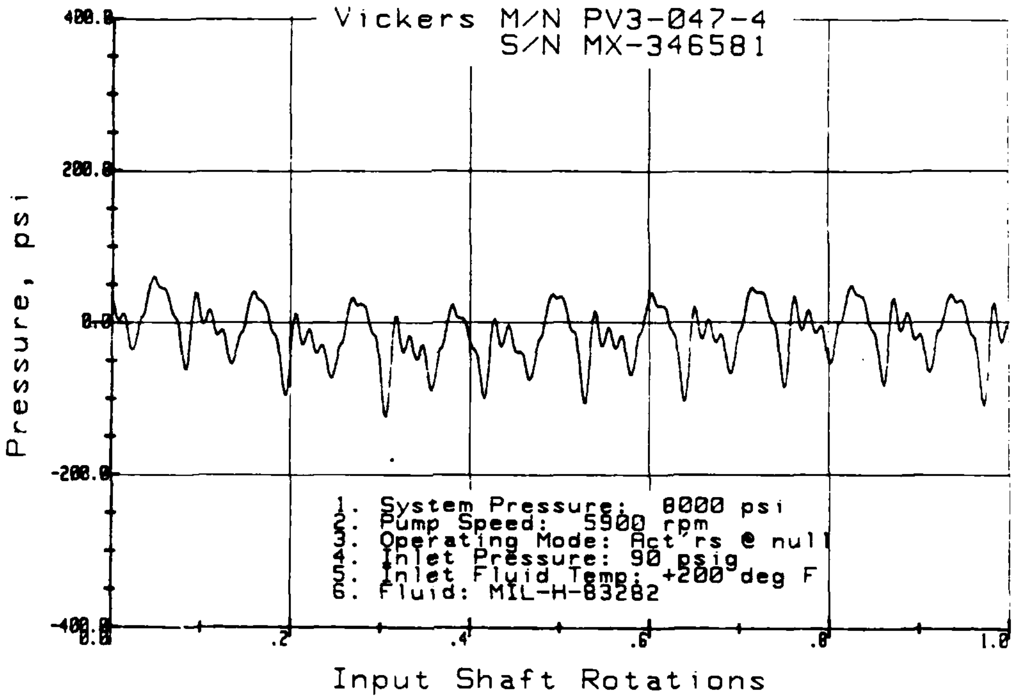
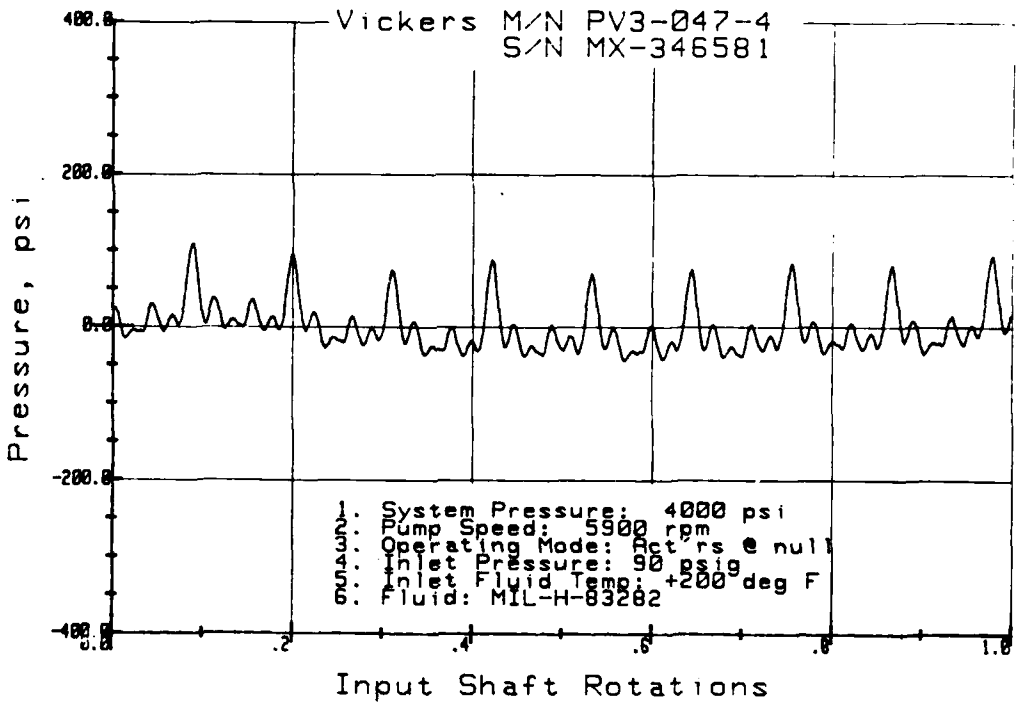




PUMP PRESSURE RIPPLE, FC-1 SYS.





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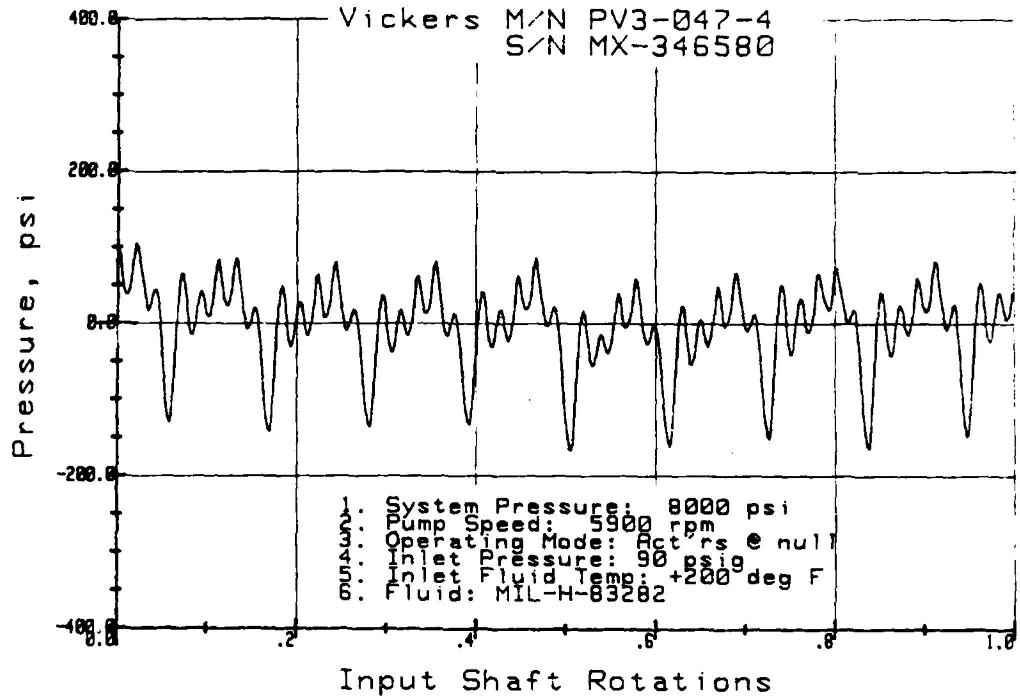
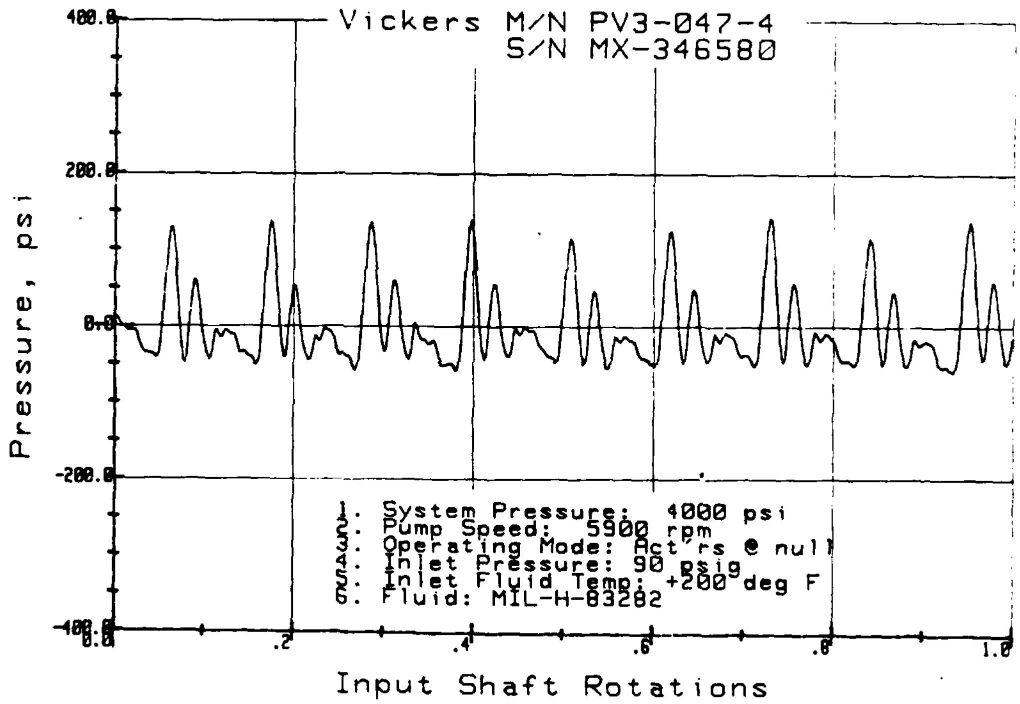
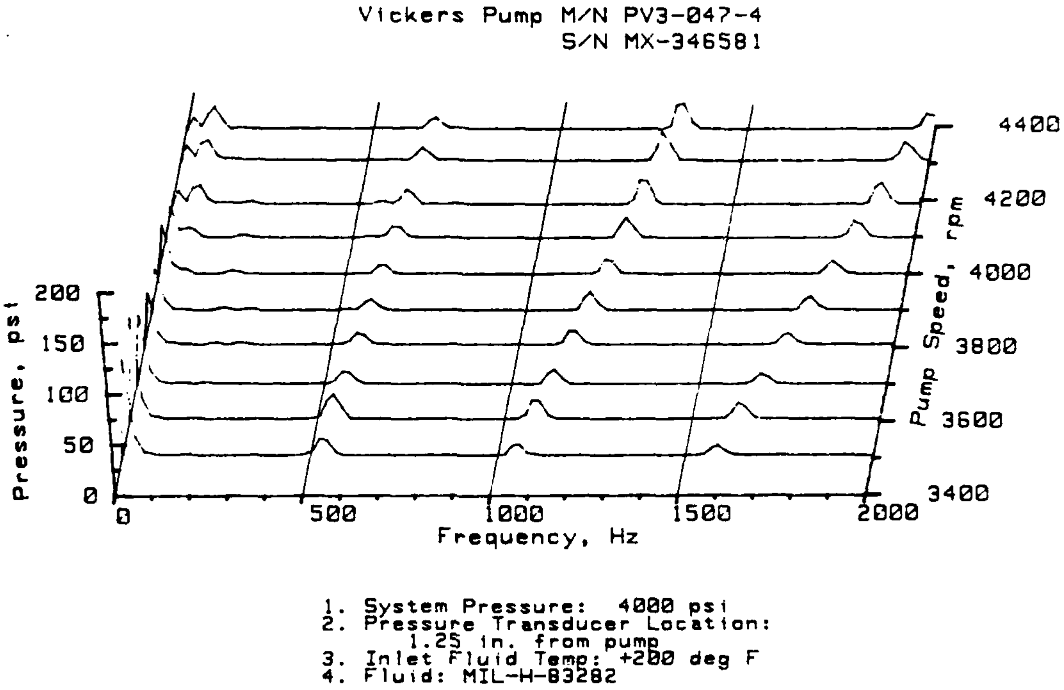
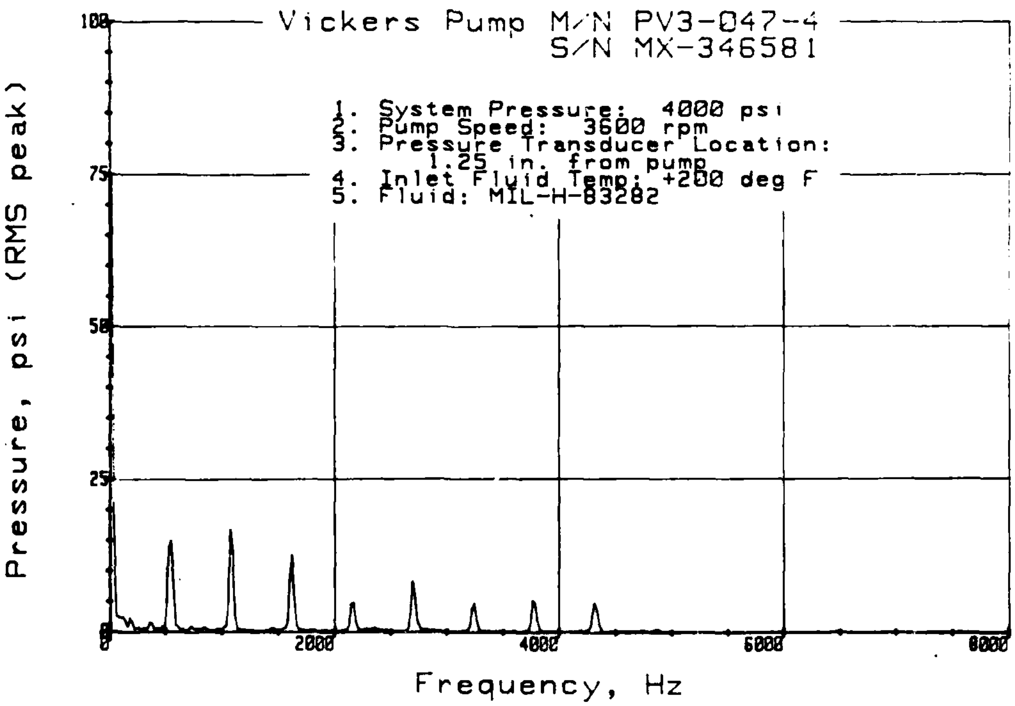


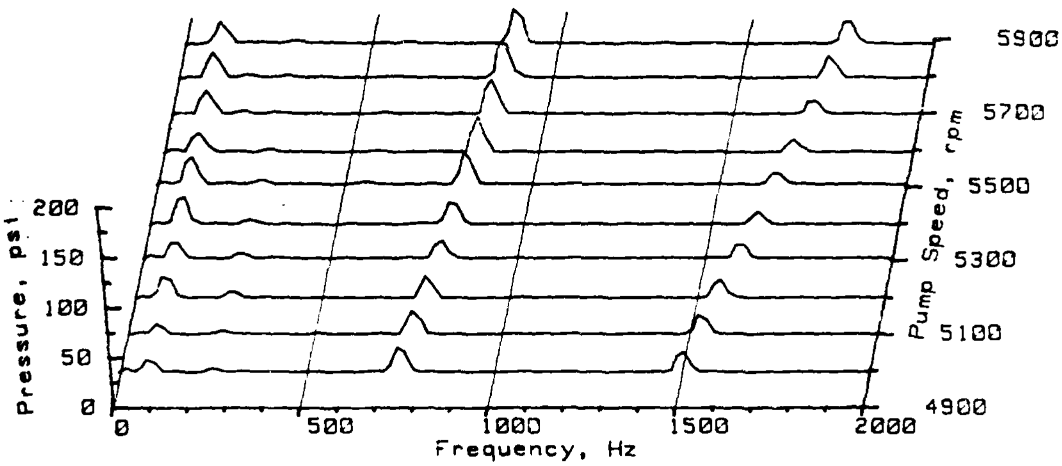


SPECTRUM ANALYSIS, FC-1 SYSTEM

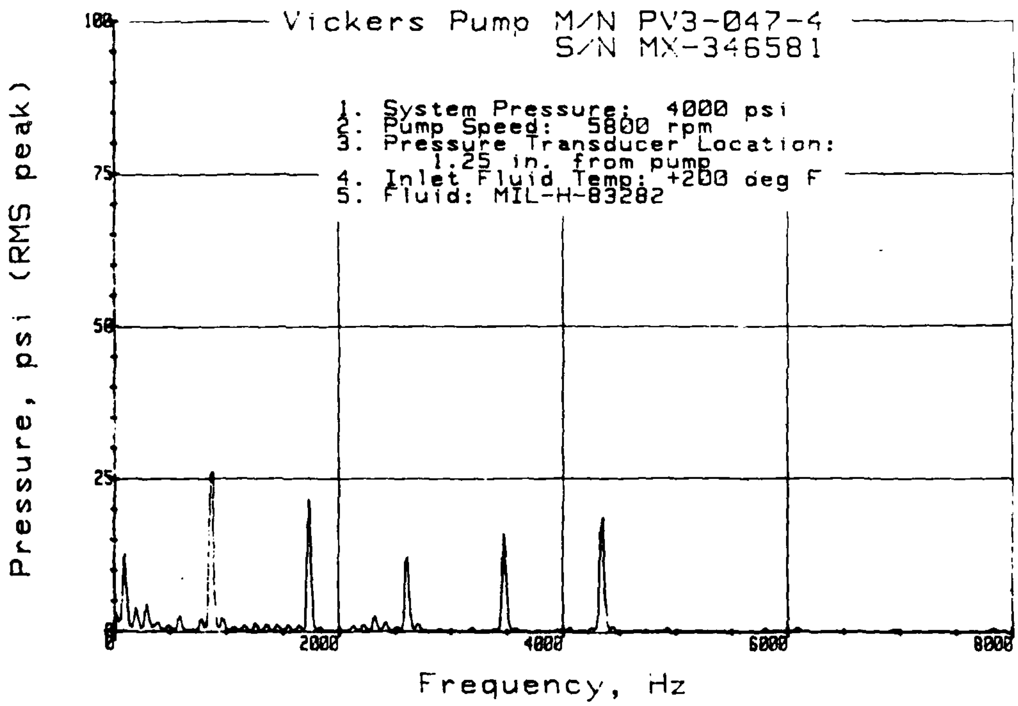




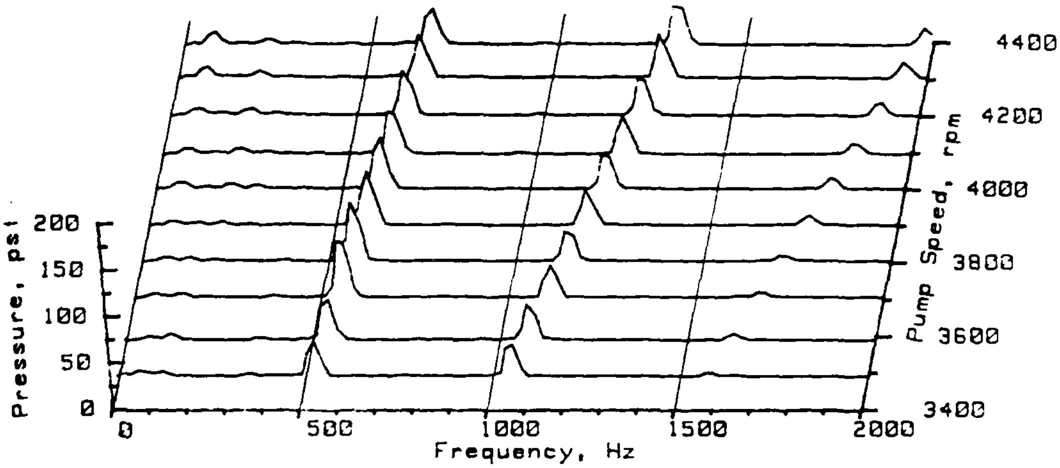
SPECTRUM ANALYSIS, FC-1 SYSTEM



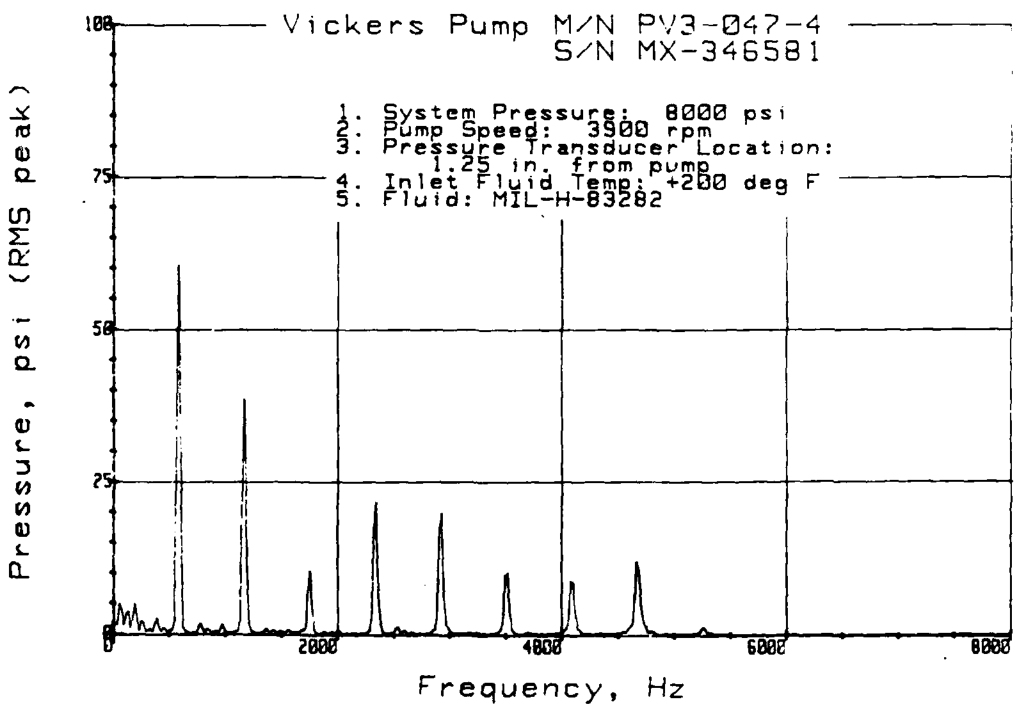
1. System Pressure: 4000 psi
2. Pressure Transducer Location: 1.25 in. from pump
3. Inlet Fluid Temp: +200 deg F
4. Fluid: MIL-H-83282



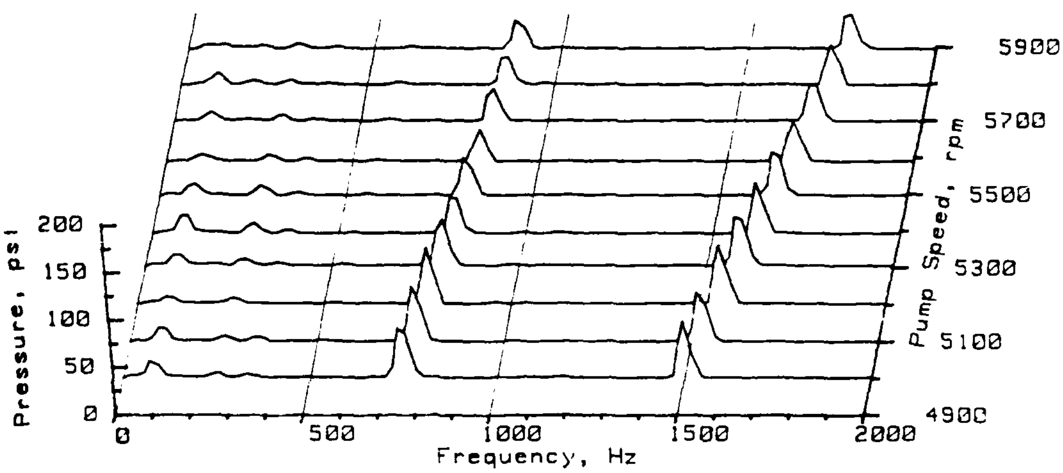
SPECTRUM ANALYSIS, FC-1 SYSTEM



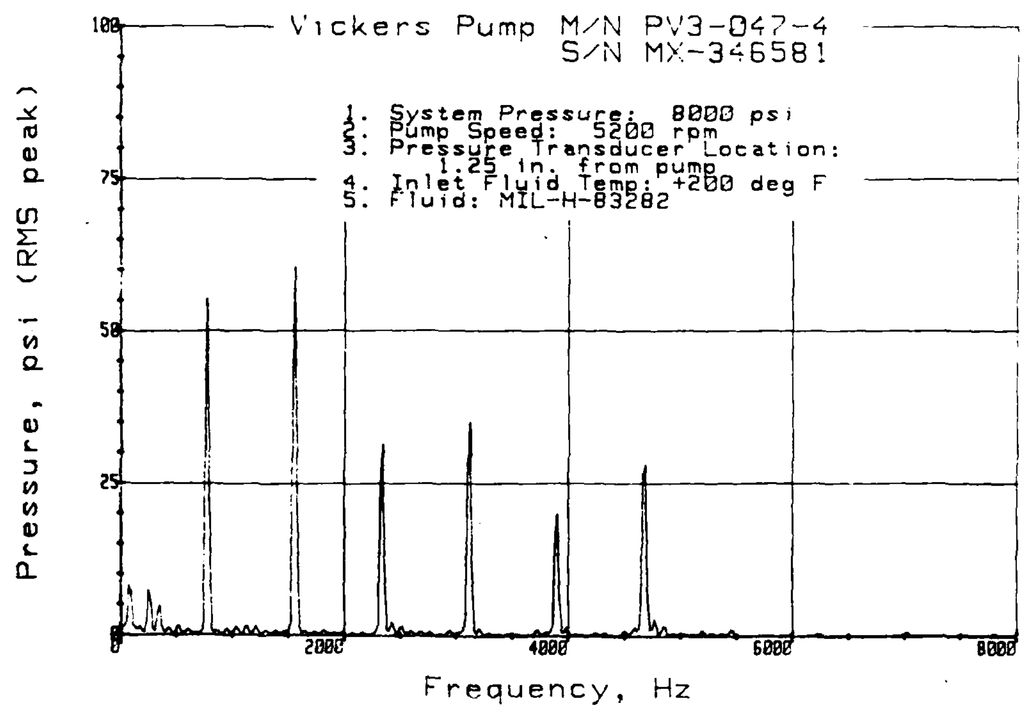
1. System Pressure: 8000 psi
2. Pressure Transducer Location: 1.25 in. from pump
3. Inlet Fluid Temp: +200 deg F
4. Fluid: MIL-H-83282



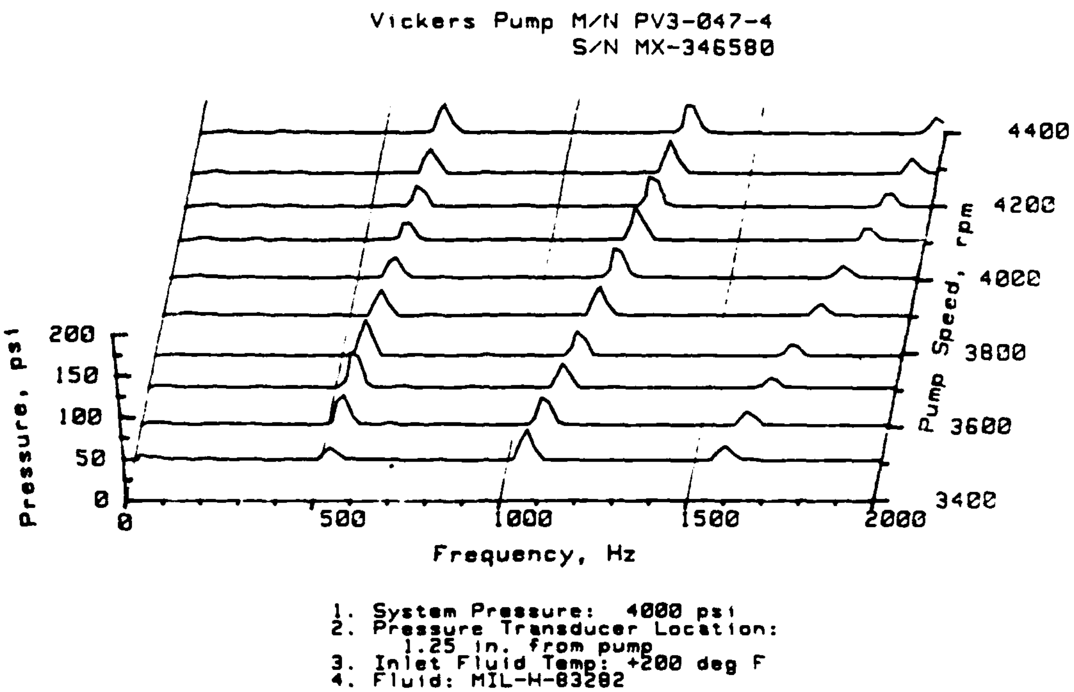
SPECTRUM ANALYSIS, FC-1 SYSTEM

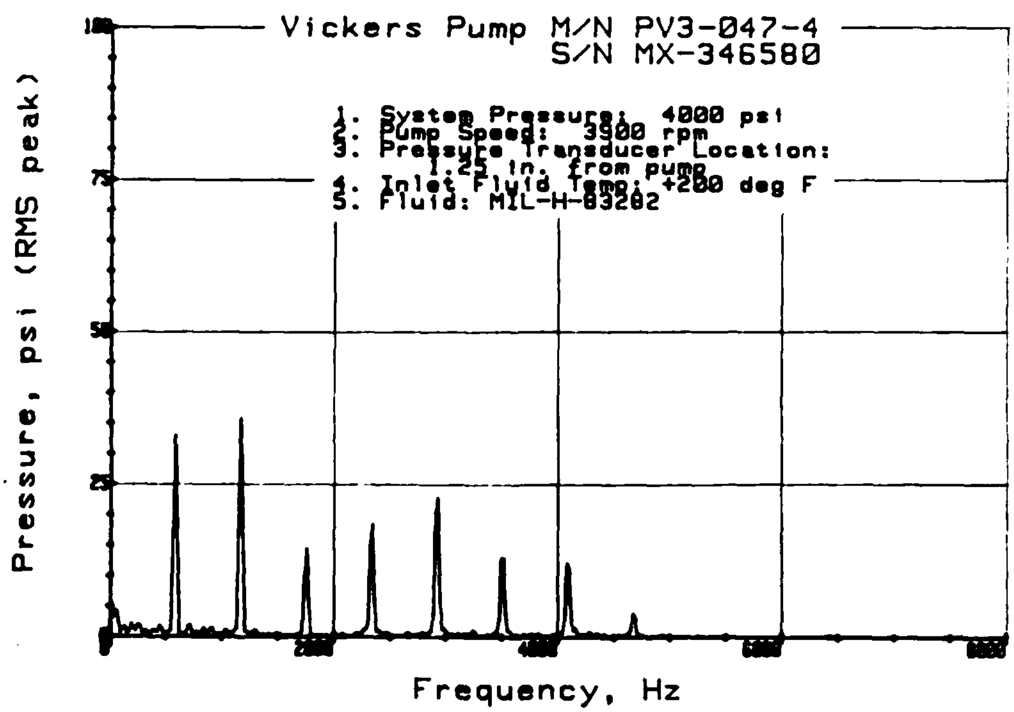


1. System Pressure: 8000 psi
2. Pressure Transducer Location: 1.25 in. from pump
3. Inlet Fluid Temp: +200 deg F
4. Fluid: MIL-H-83282



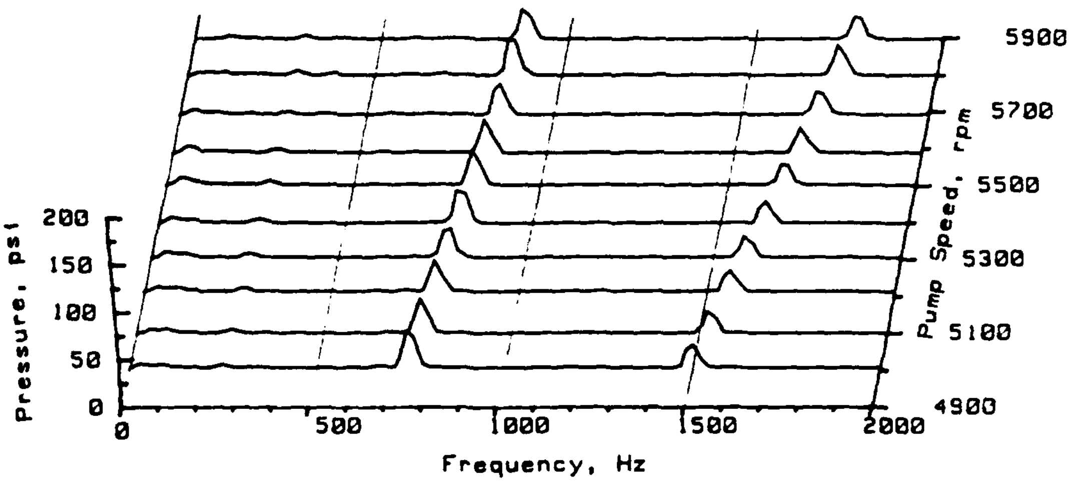
SPECTRUM ANALYSIS, FC-2 SYSTEM



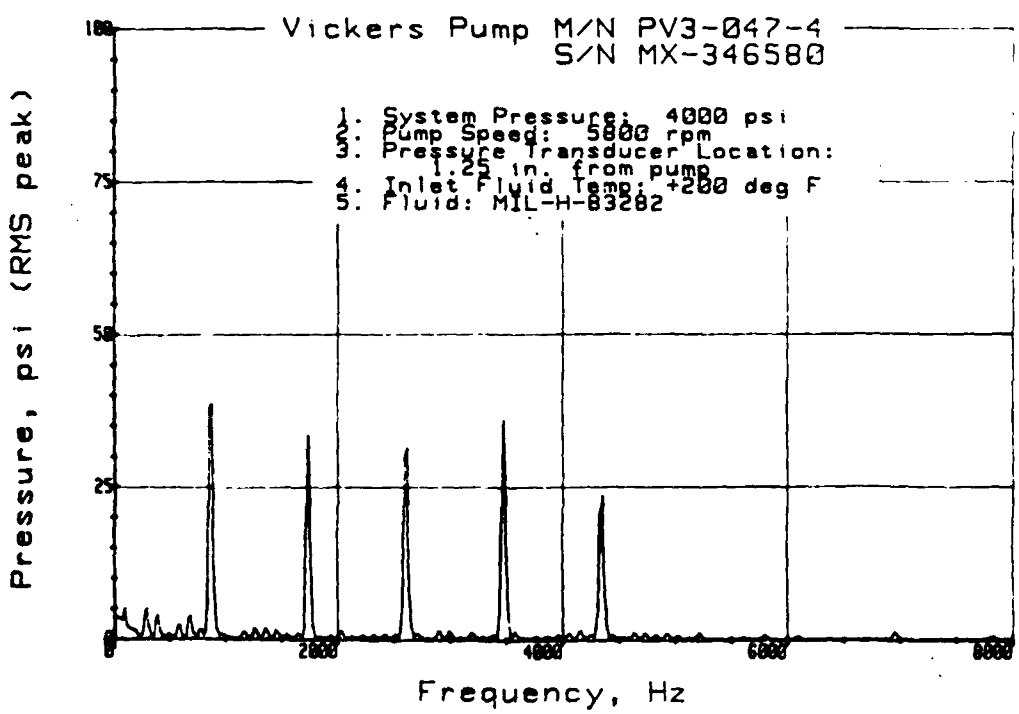


SPECTRUM ANALYSIS, FC-2 SYSTEM

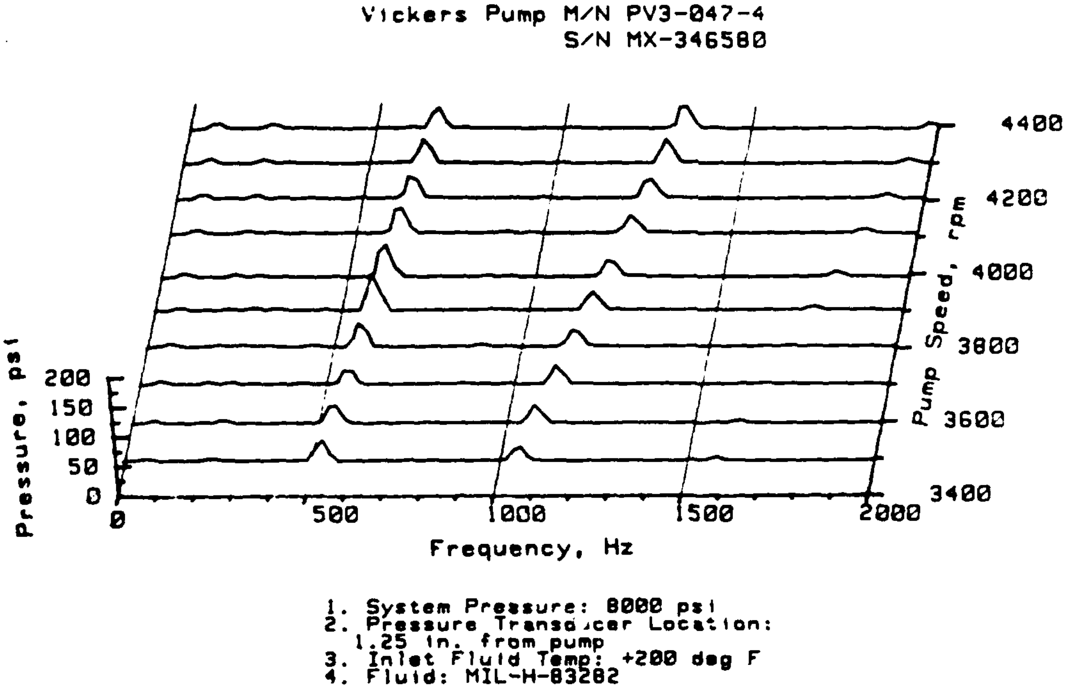
Vickers Pump M/N PV3-047-4
S/N MX-346580

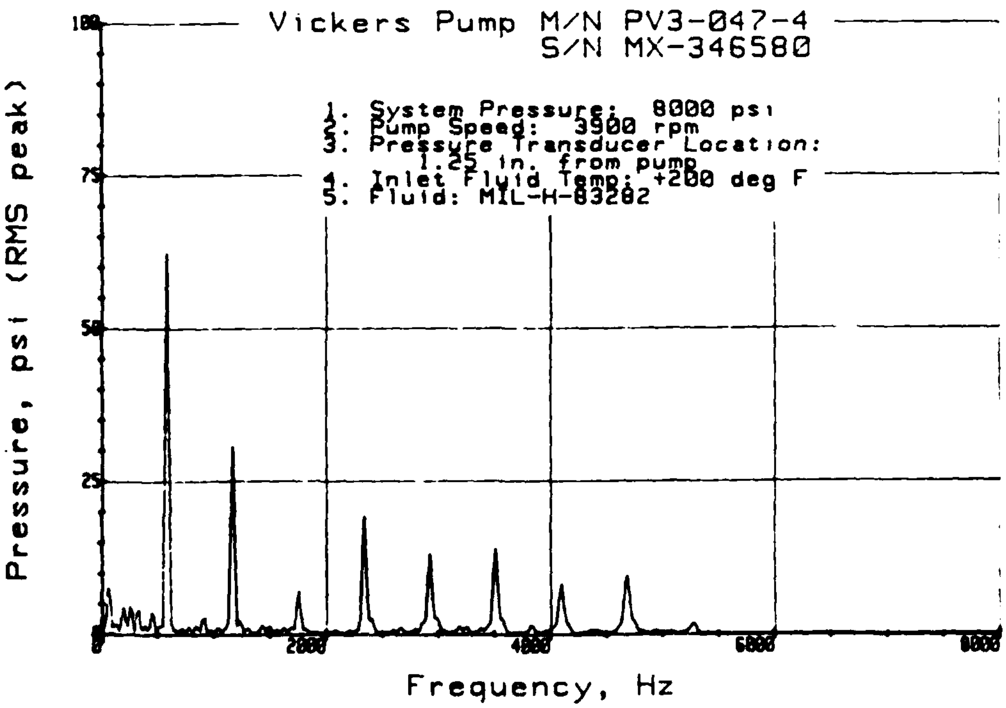


1. System Pressure: 4000 psi
2. Pressure Transducer Location: 1.25 in. from pump
3. Inlet Fluid Temp: +200 deg F
4. Fluid: MIL-H-83282

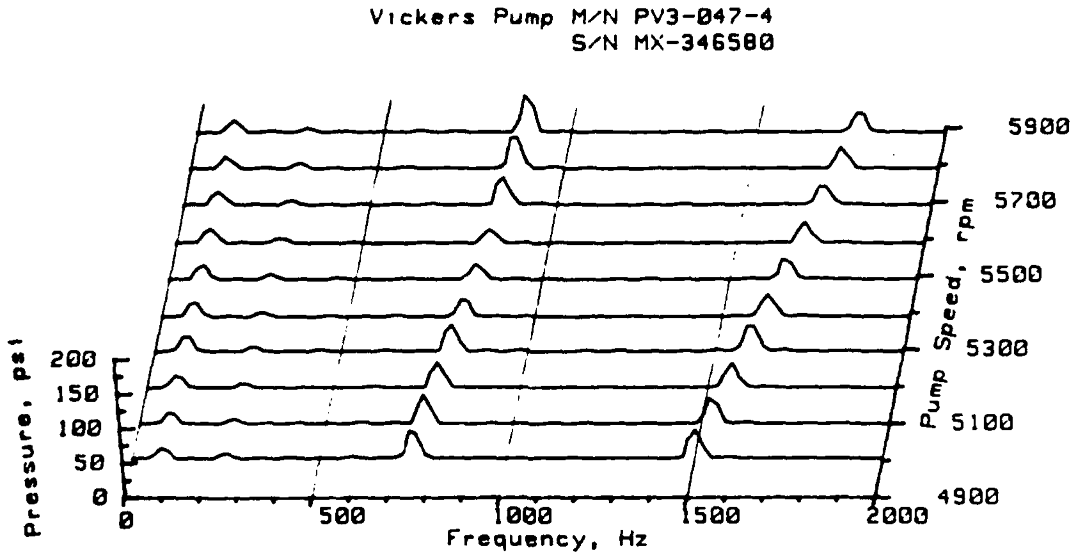


SPECTRUM ANALYSIS, FC-2 SYSTEM

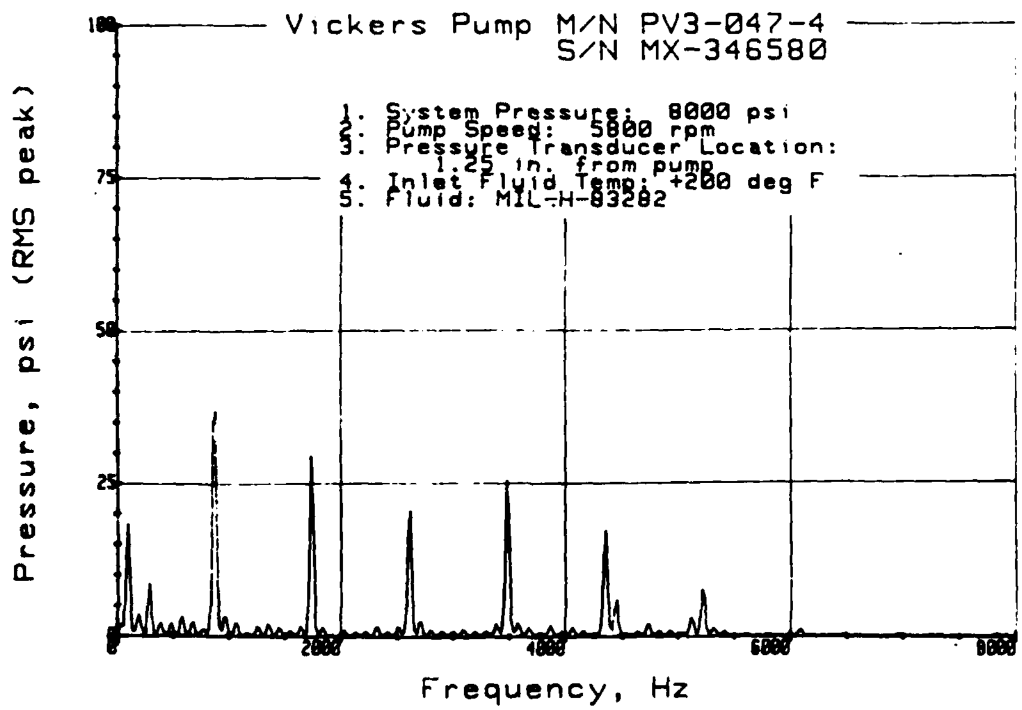




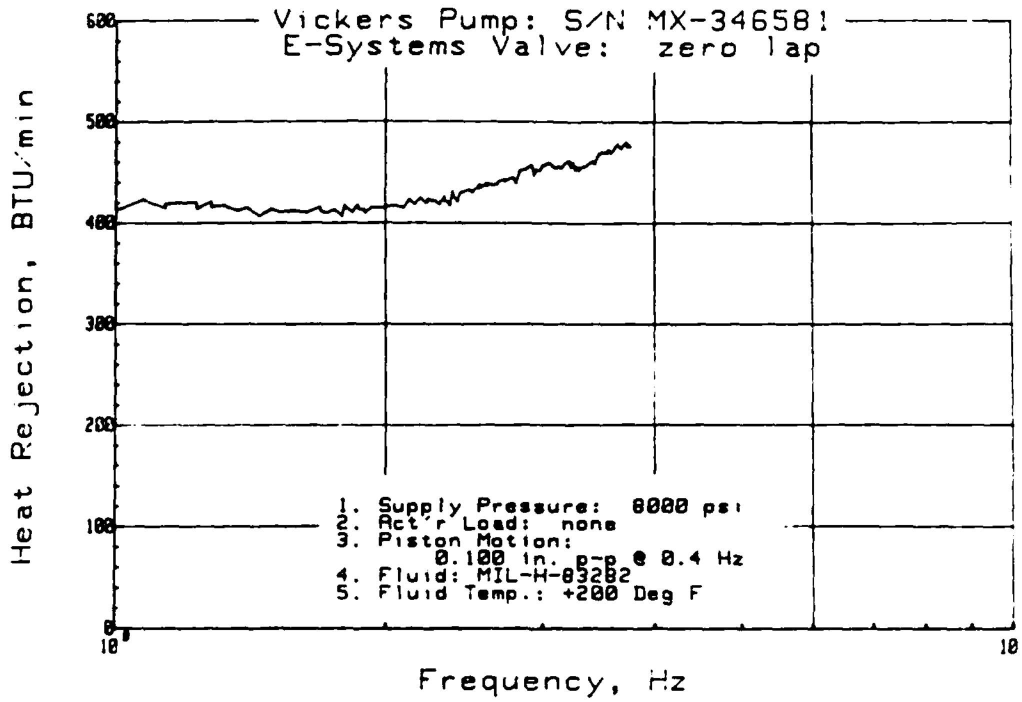
SPECTRUM ANALYSIS, FC-2 SYSTEM

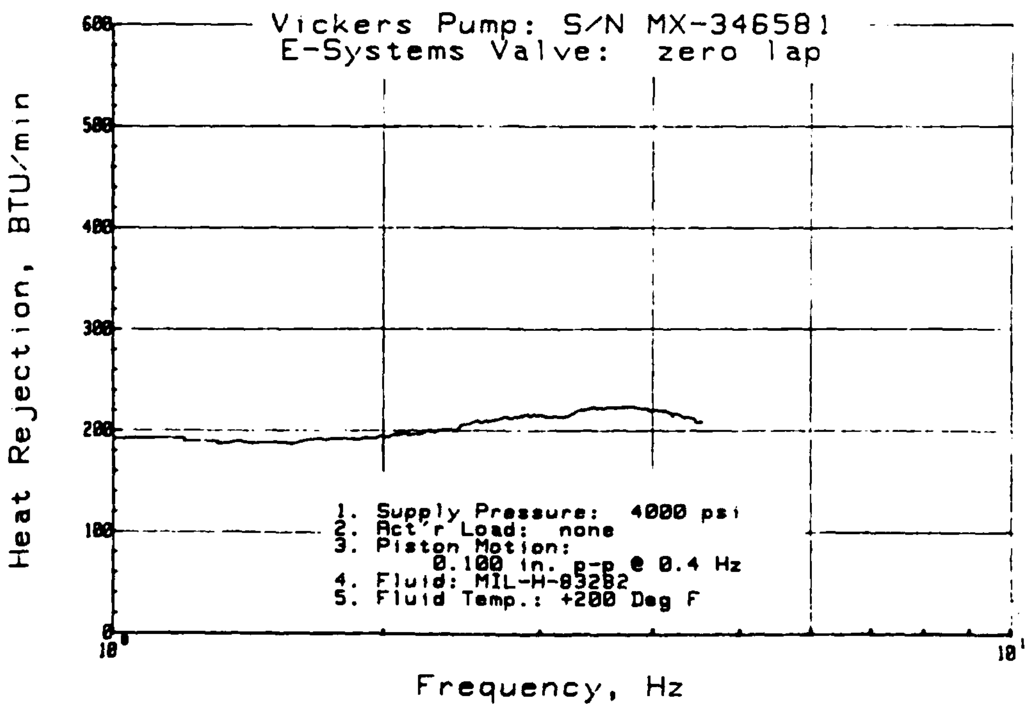


1. System Pressure: 8000 psi
2. Pressure Transducer Location: 1.25 in. from pump
3. Inlet Fluid Temp: +200 deg F
4. Fluid: MIL-H-83282

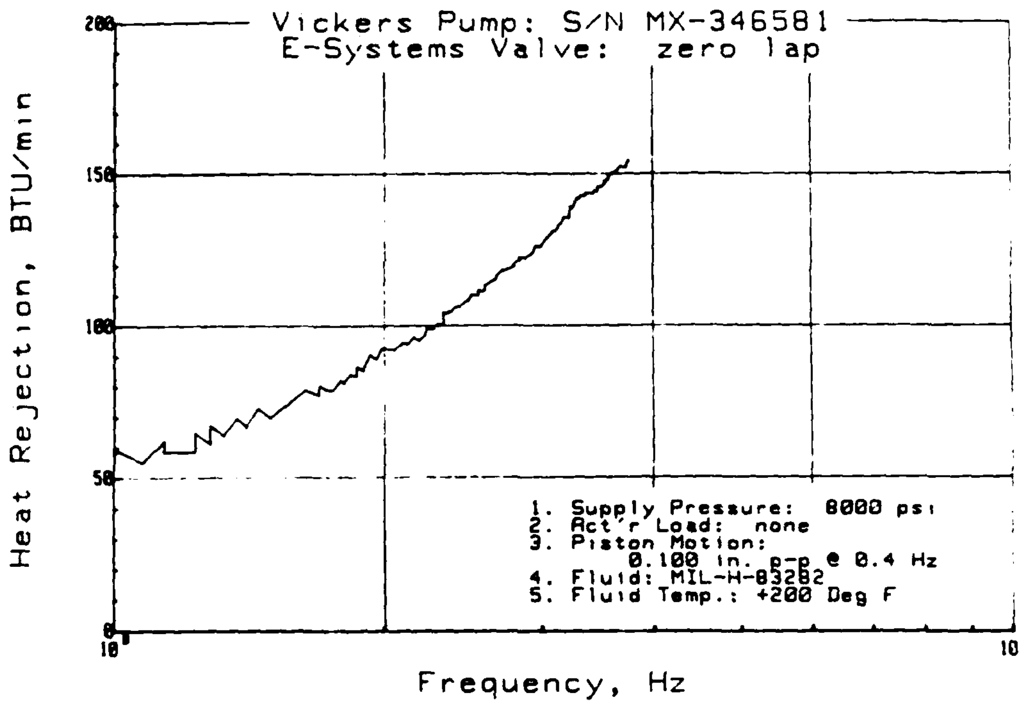


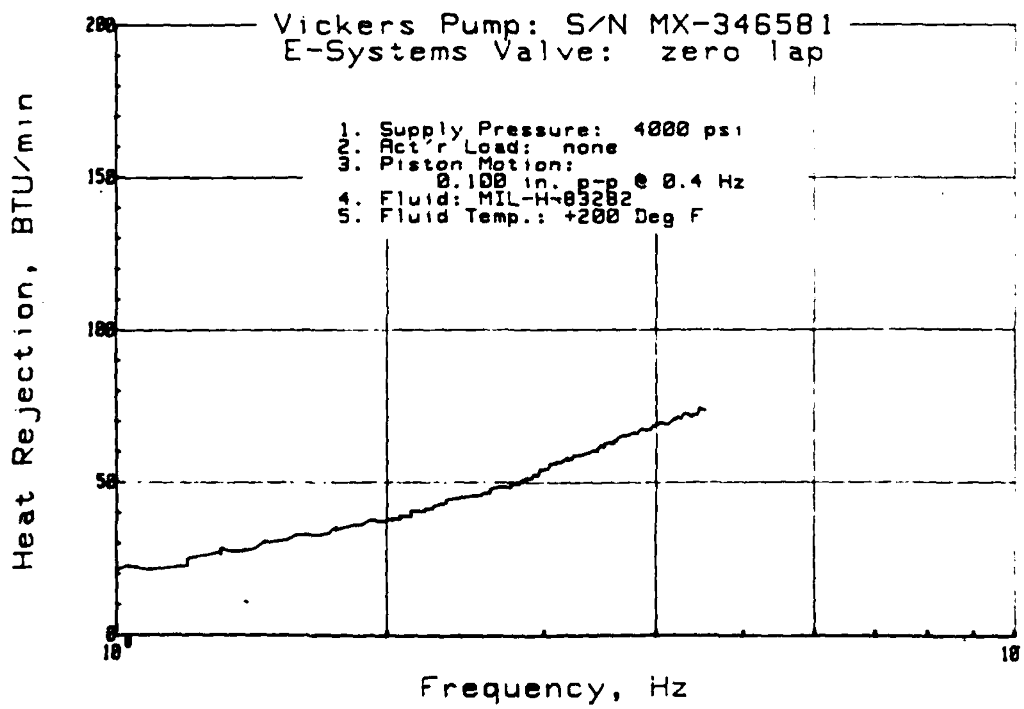
PUMP ENERGY CONSUMPTION



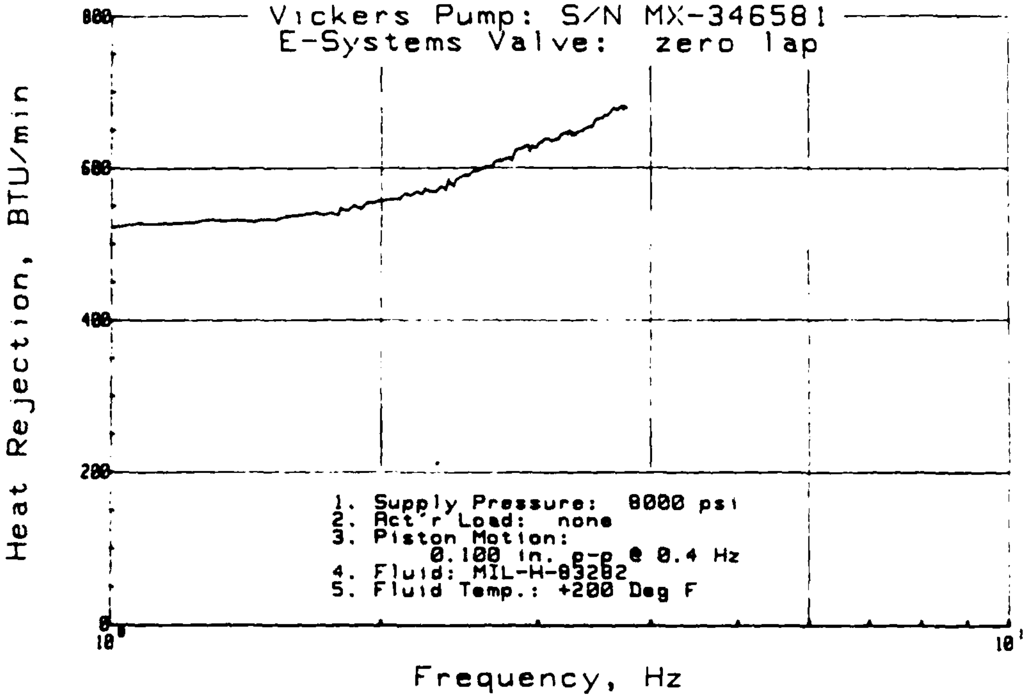


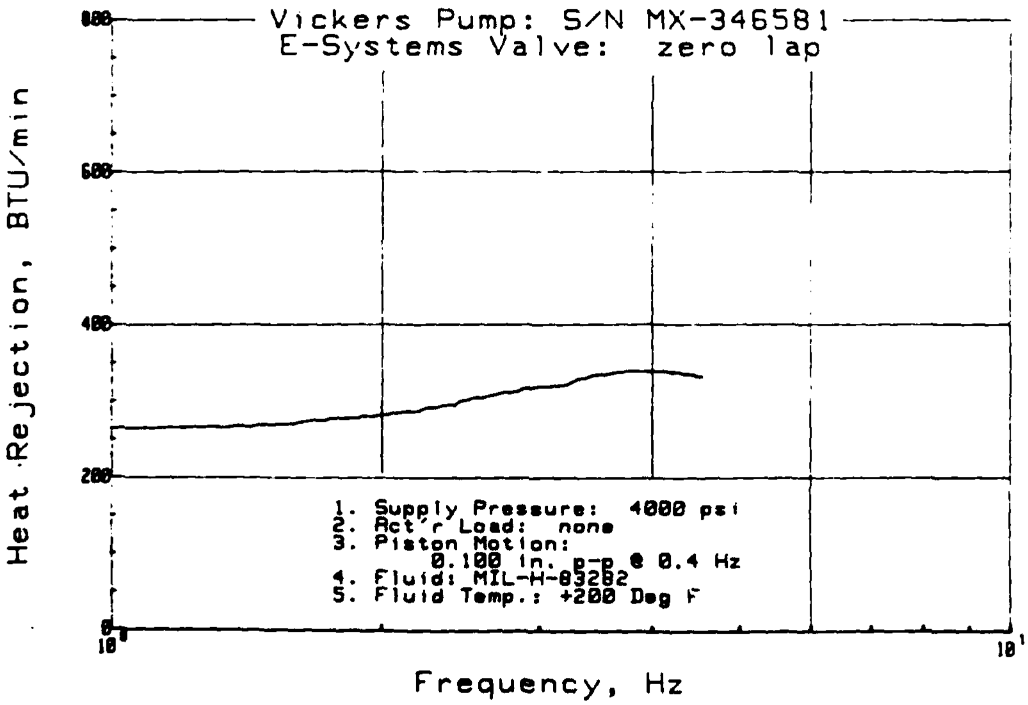
ACTUATOR ENERGY CONSUMPTION





SYSTEM ENERGY CONSUMPTION





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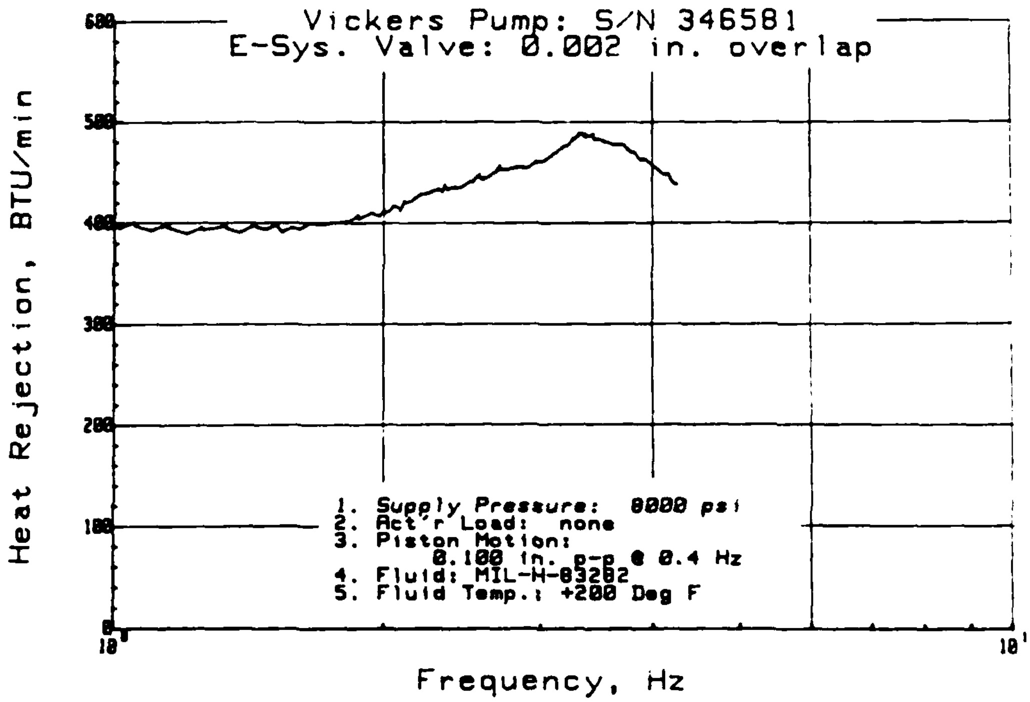

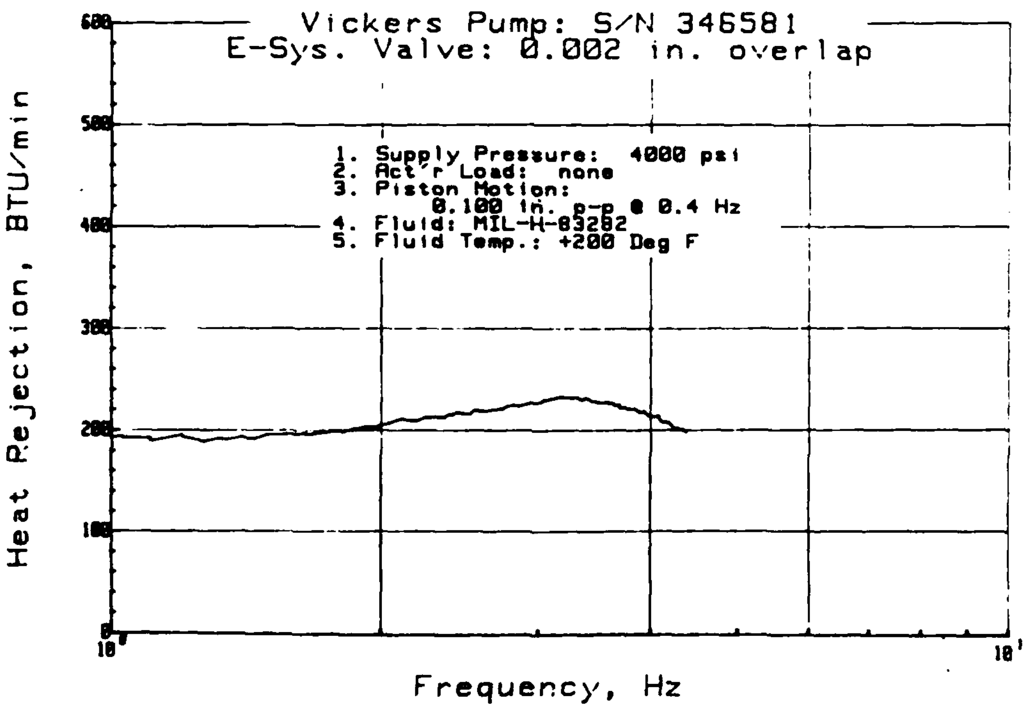

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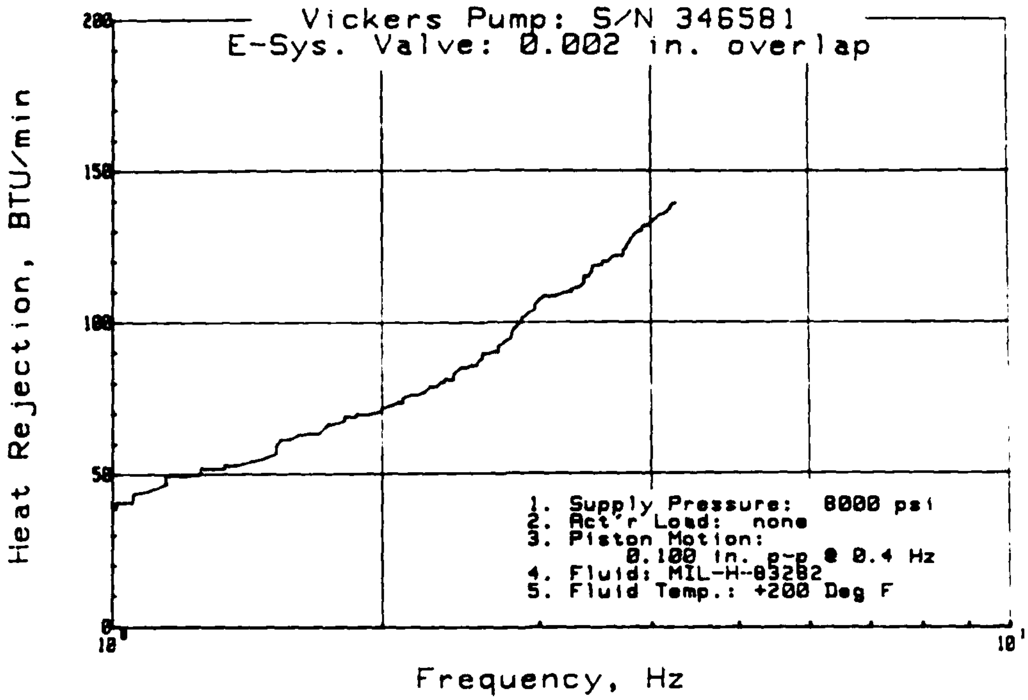

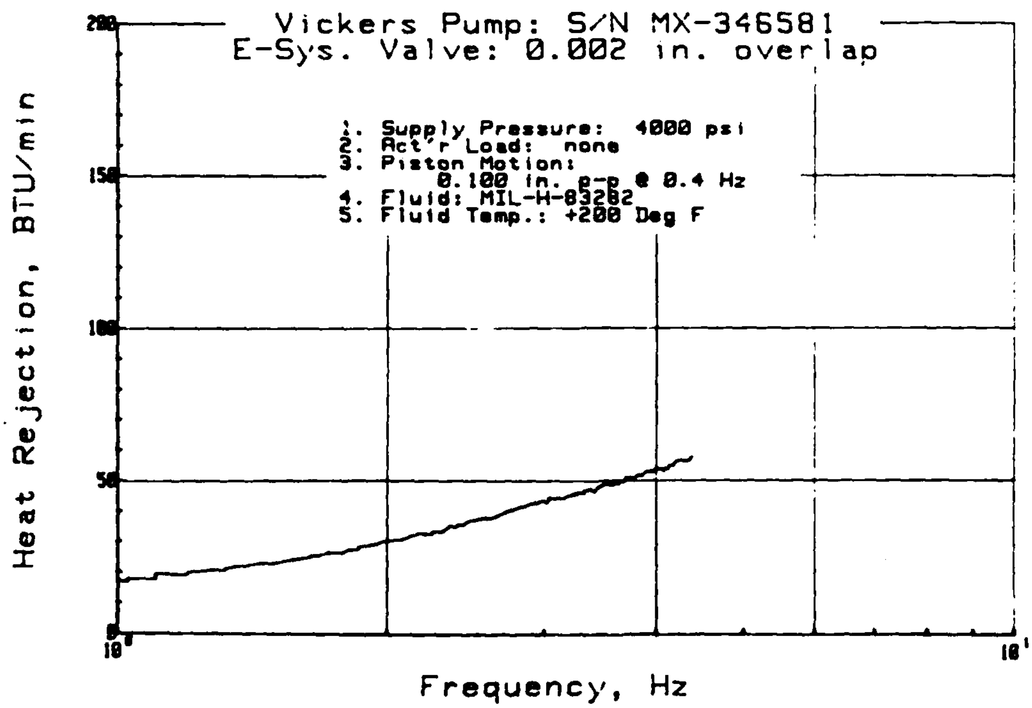

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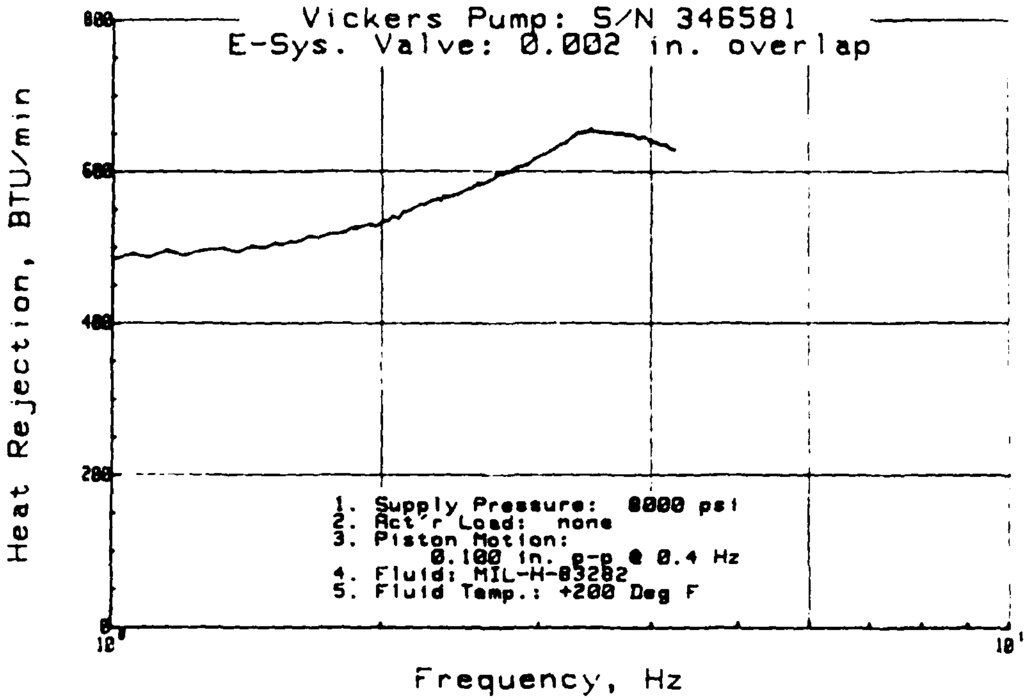

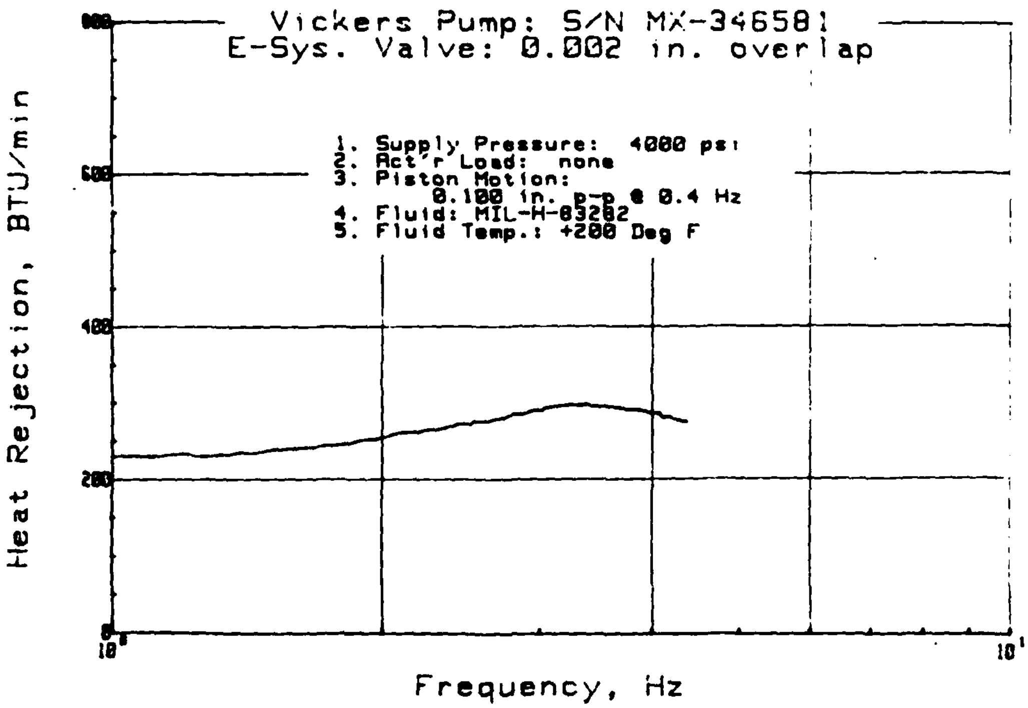

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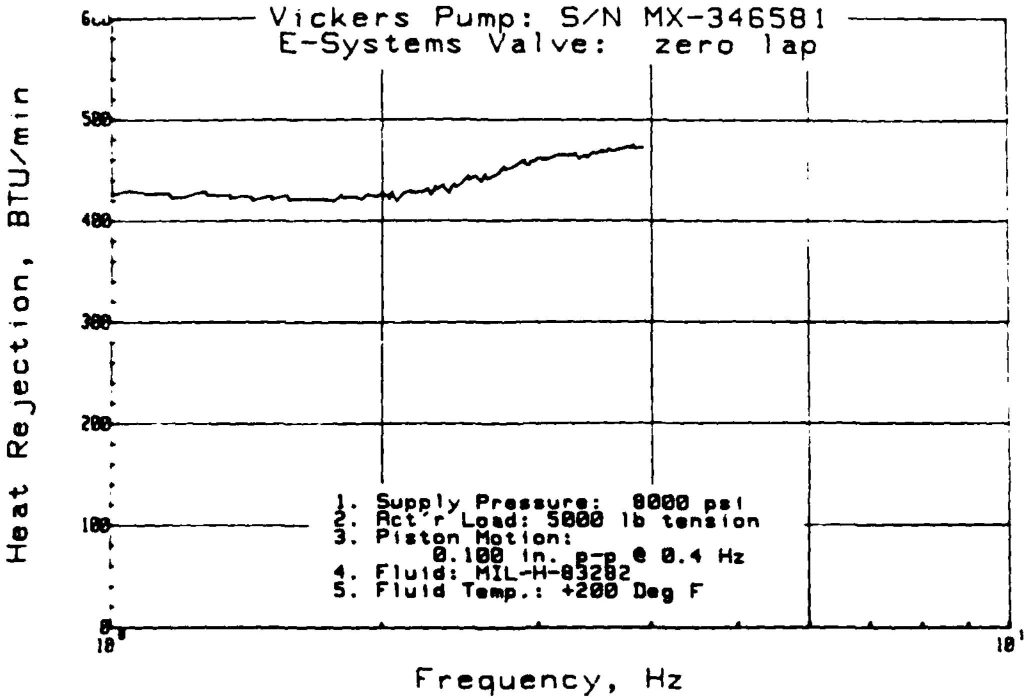

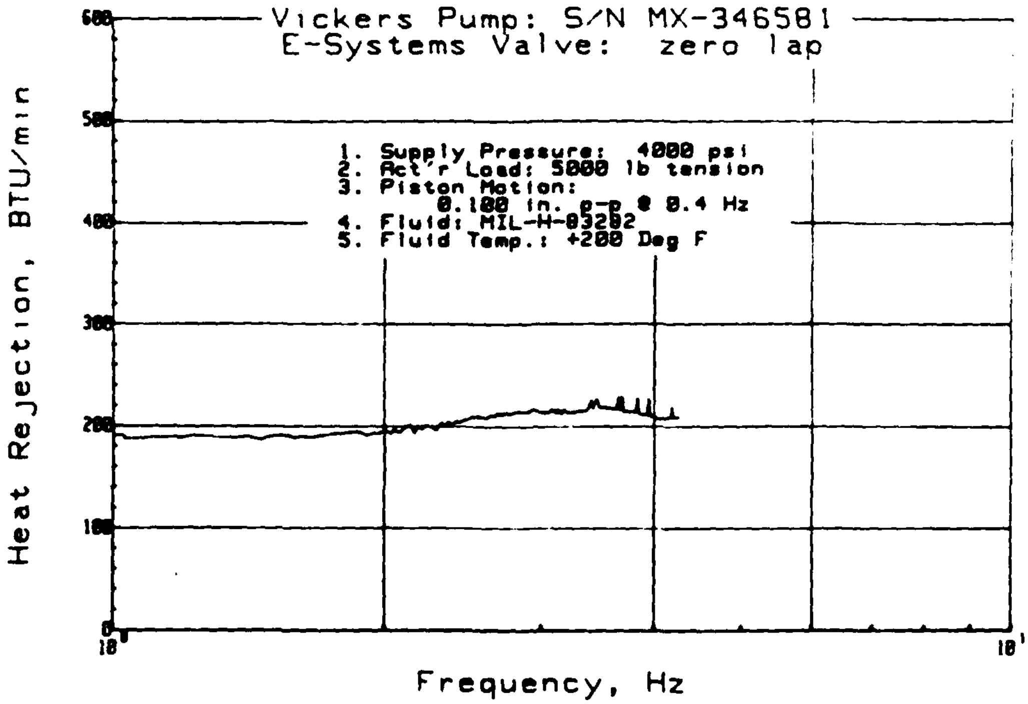

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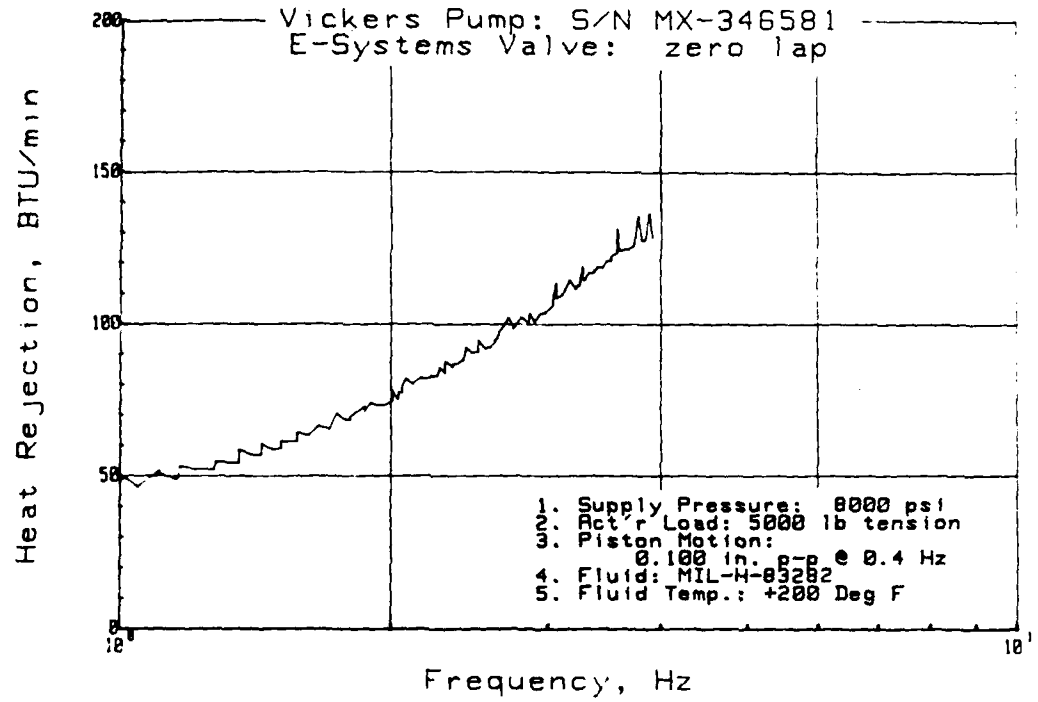
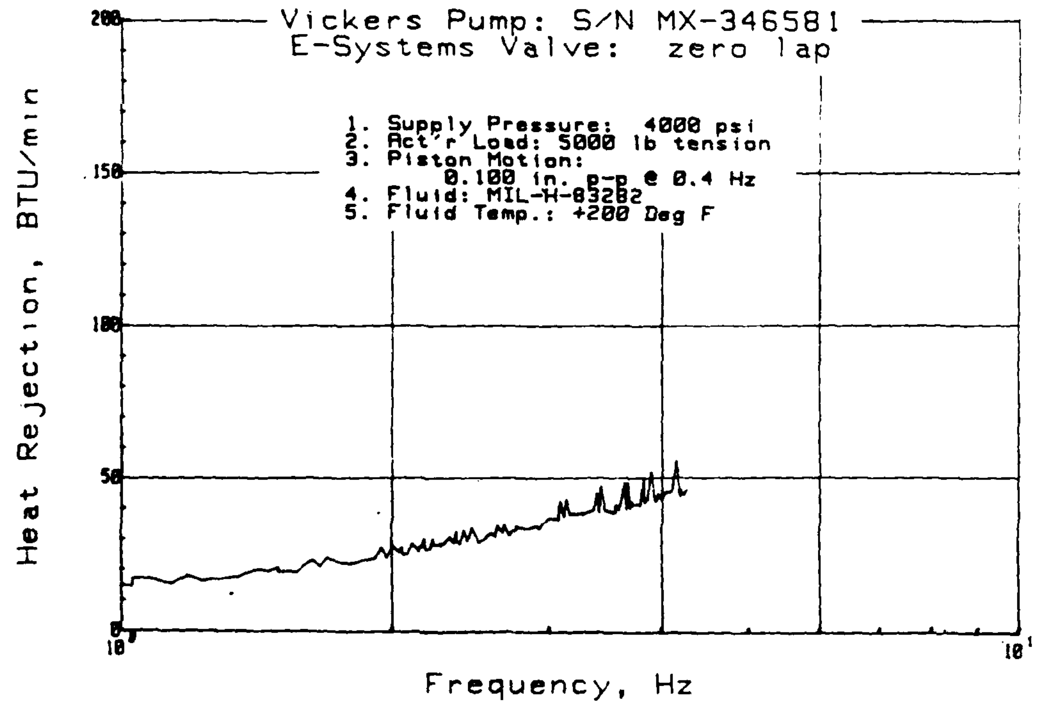
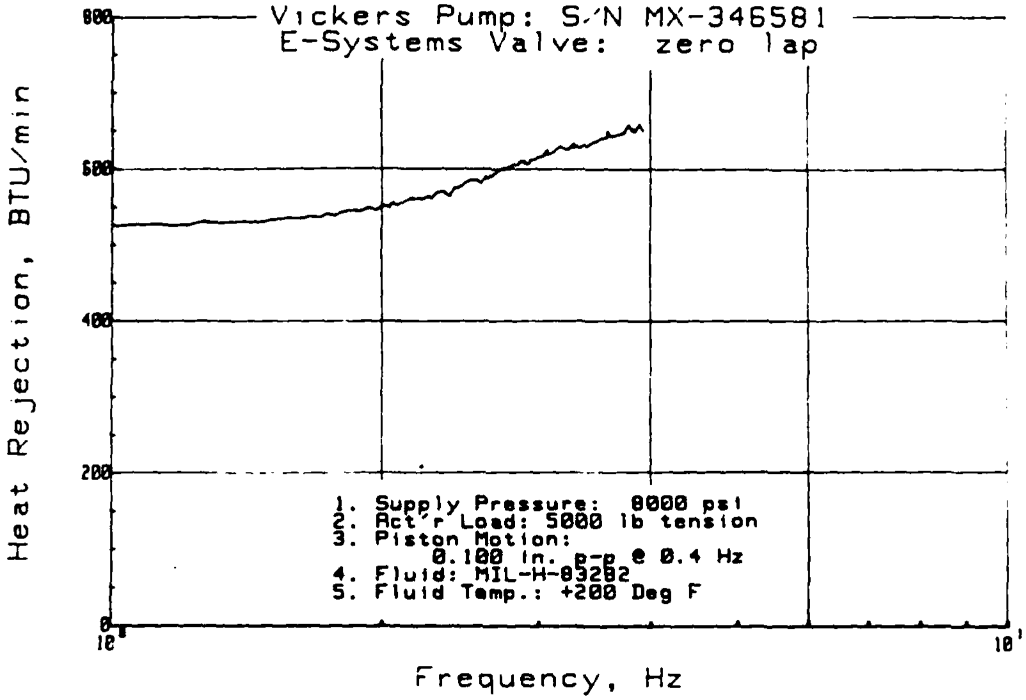
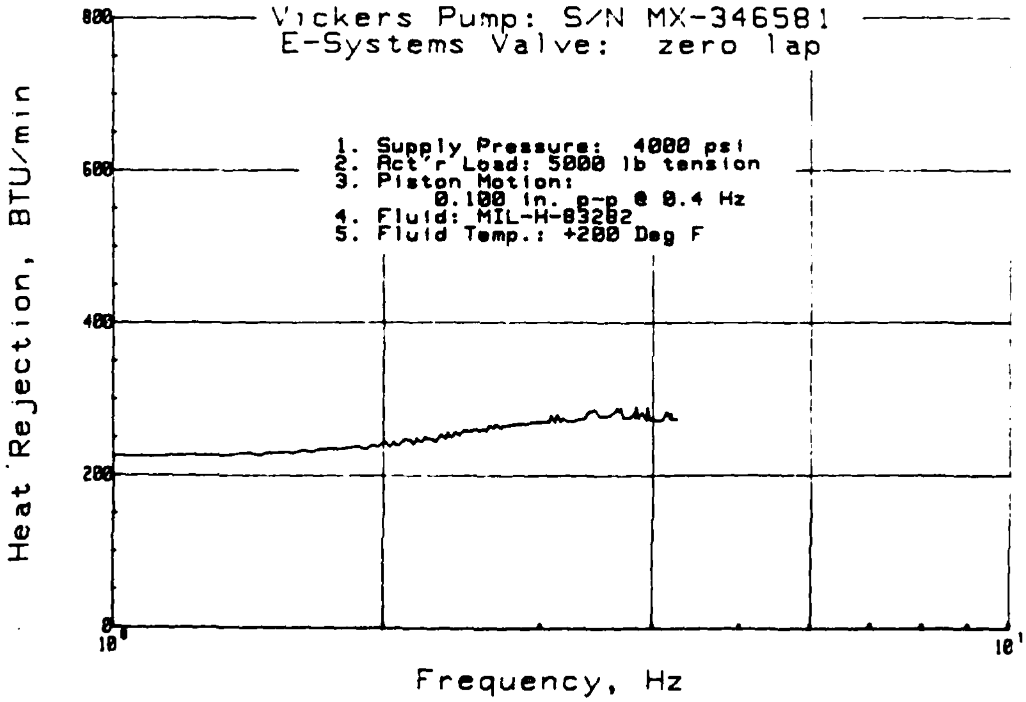




SYSTEM ENERGY CONSUMPTION





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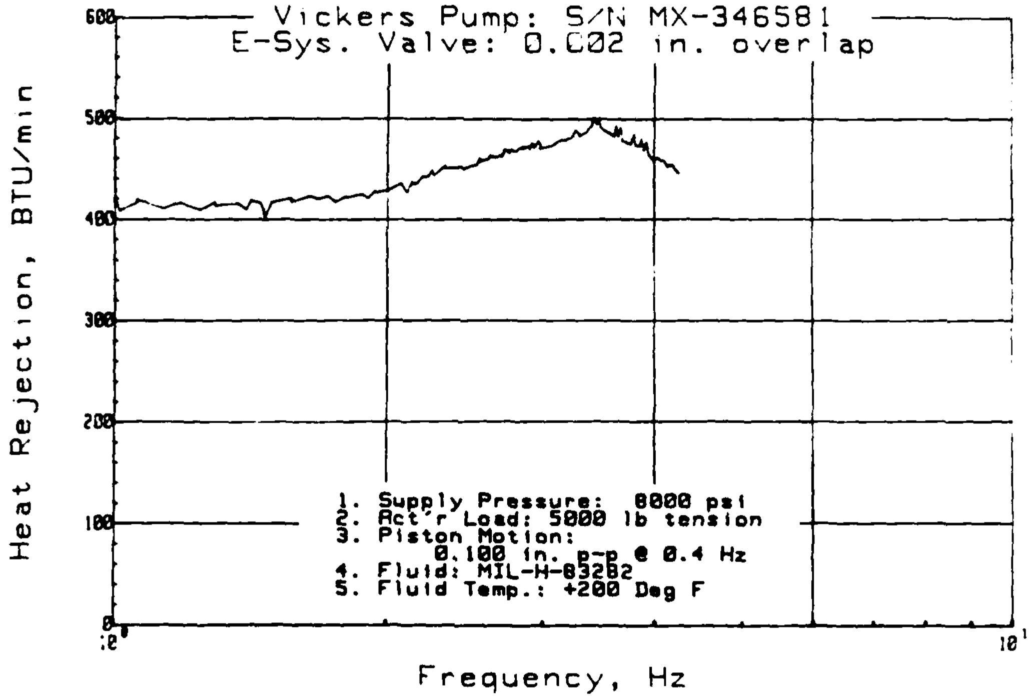

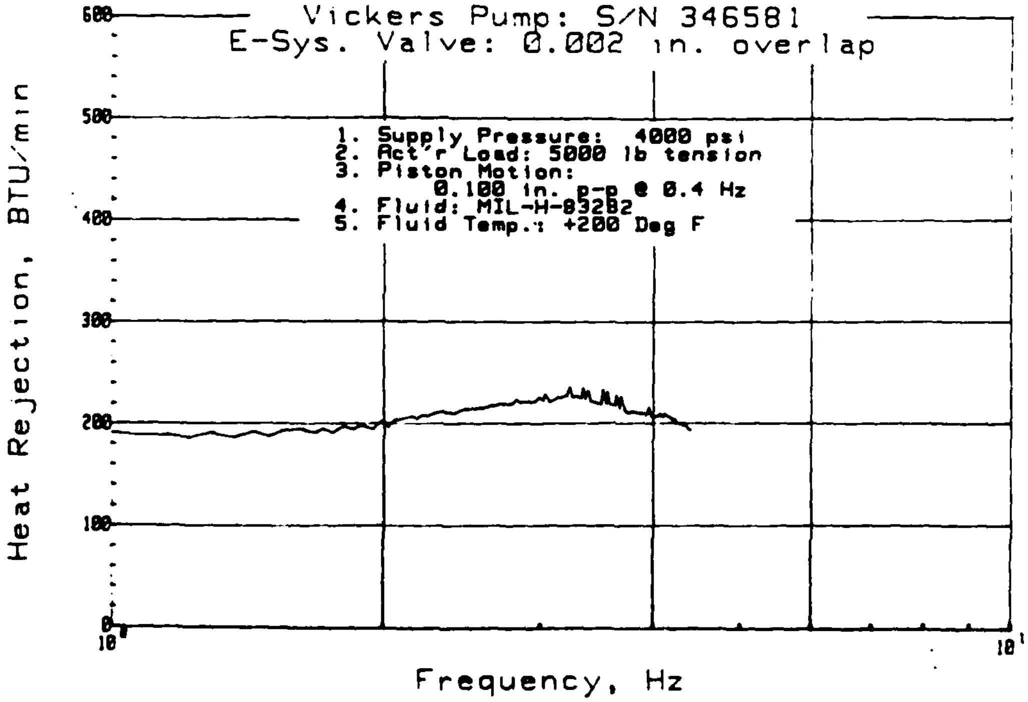

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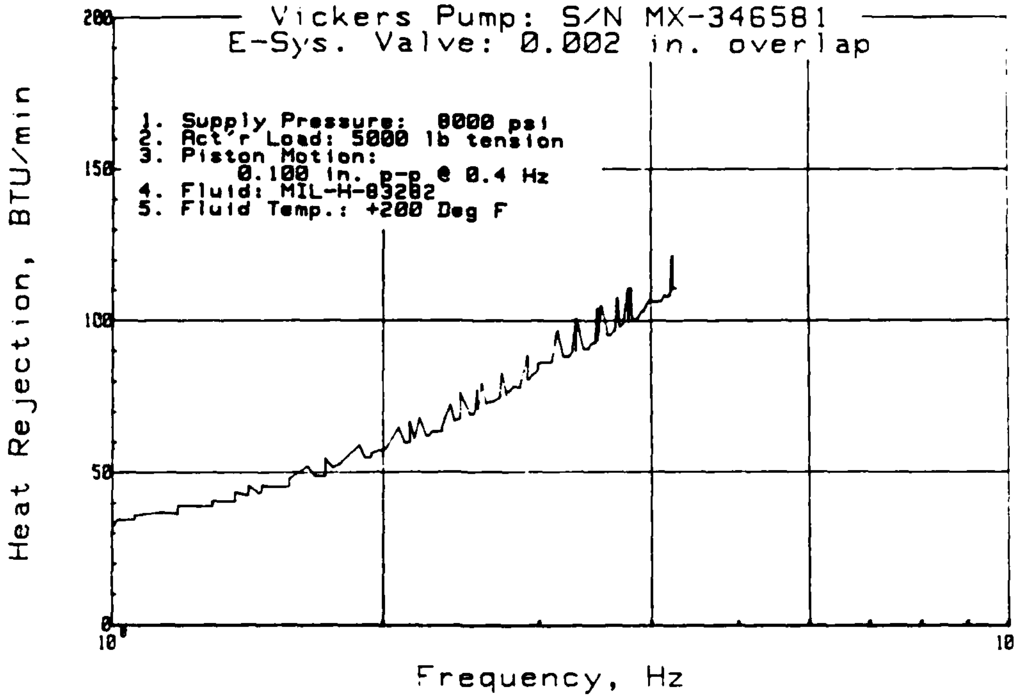

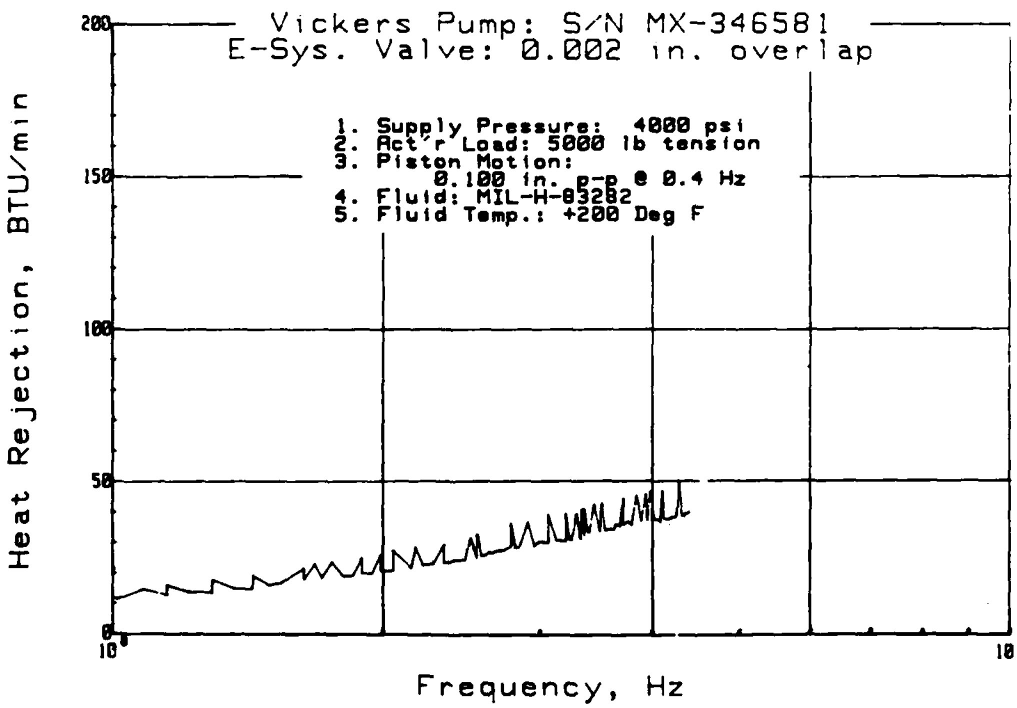

SYSTEM ENERGY CONSUMPTION